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TRG'S POSITION PAPER ON THE CONFORMAL/PLANAR ARRAY DESIGN.(U)

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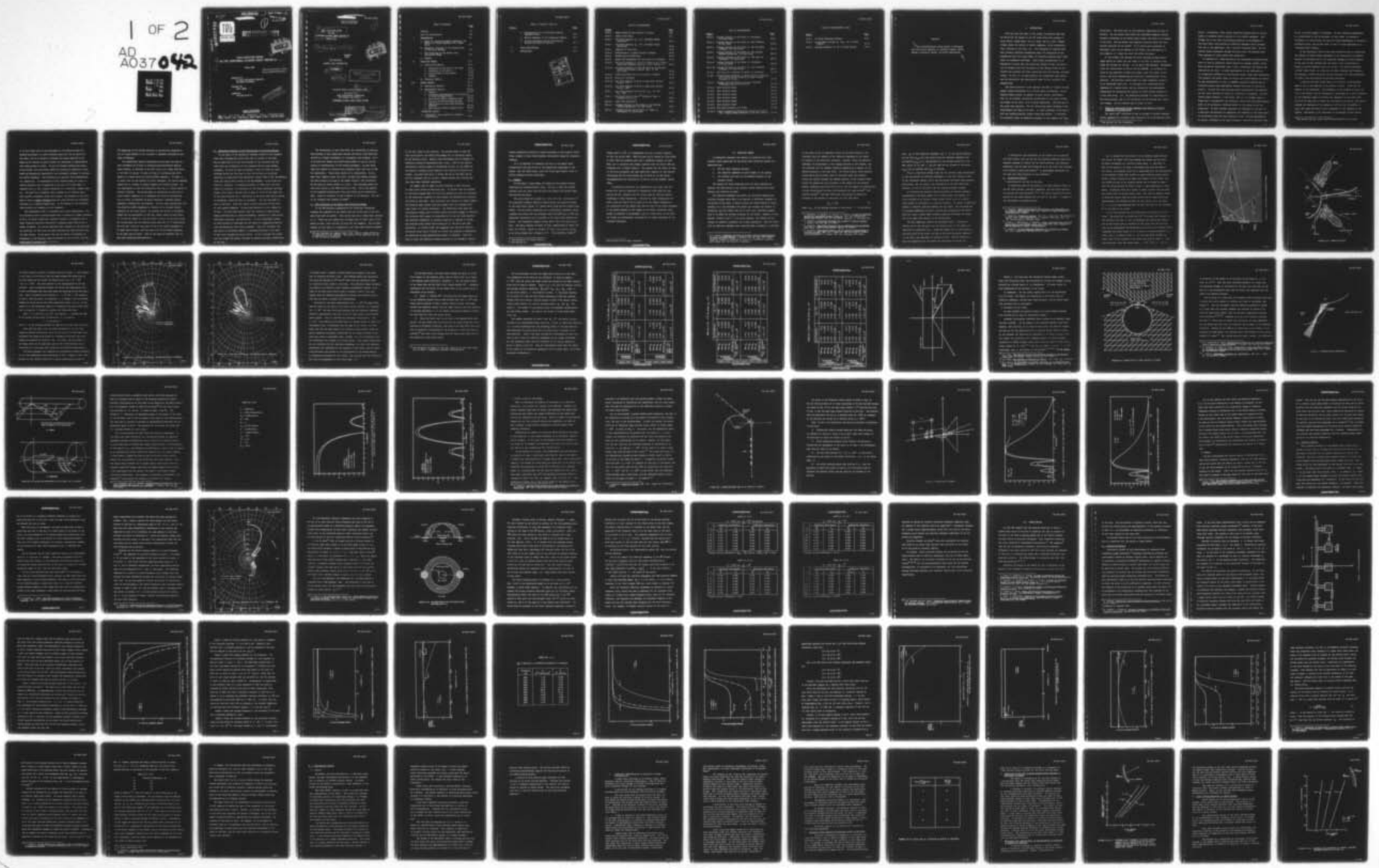

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TRG'S POSITION PAPER ON THE CONFORMAL/PLANAR ARRAY DESIGN (U)

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CONFORMAL/PLANAR ARRAY SONAR SYSTEM

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Abstract

The conformal/planar array design is discussed with particular emphasis on radiation theory, baffle design, transducer design, and naval engineering.

I. INTRODUCTION

Over the last few years a wide range of proposals have been put forth for the design of the C/P sonar array with respect to array factor, wedge angle, use of a dome, use of a boot, element design, means for control of baffle impedance, array orientation, size, location on the ship, etc. The divergence of opinion probably reflects different assumptions about the objectives of the program and also the large number of technical problems about which there is inadequate knowledge. Under these circumstances it is impossible to prove that any particular design is best; one has to try to estimate what will be gained if each design should be successful and estimate the risks associated with that design, and then choose. We will try to make explicit our assumptions about goals, and the theoretical and experimental bases for our technical recommendations.

The prime objective in our approach has been to realize as much weapon system performance for as little money as possible. We emphasize weapon system, as opposed to sonar system, performance. That is, we consider the penalty attributable to the sonar in drag and weight on the ship, to be of great importance. This has led to the sonar keel approach. The air filled keel adds a minimum to the displacement and drag of the ship. It takes advantage of the ship's hull for baffling against surface noise and bubbles. It provides an accessible space in immediate proximity to the elements for sonar

electronics. The sonar keel is load bearing, supporting the ship in drydock. The sea chests which house the transducer elements provide resonant stiffening of the baffle which permits the array to be steered to end fire. The sea chests also suppress structure borne and flow excited vibration of the baffle. If it should prove necessary or desirable a boot can be adhered to the baffle, and individually to the elements, or a dome can cover the entire structure.

At the present time designs for array housings having various wedge angles are under test and there is no point in trying to prejudge the outcome any further. In our March 1966 document, "Recommendations for Installation of ESS on the USS SPOKANE," we outlined our views on the questions of dome vs no-dome, array tilt angle, array factor, and naval engineering and structural considerations in our approach. The principal issues which are under immediate scrutiny, as we understand them, are: (1) The means for increasing baffle impedance in a sparse array, and the theoretical and experimental foundations for estimating the effects of finite baffle rigidity on a full size array. (2) The mechanical problems of construction of the array housing, and (3) the protection of the housing from cavitation damage. We will discuss each of these in turn.

A. Means for Increasing Baffle Impedance and Effects of Finite Impedance in a Large Array

Our early work* convinced us that an attempt to provide adequate baffle impedance by increased plate thickness led to excessively heavy

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* See Section III for references.

plates. Furthermore, these plates supported bending waves at certain angles of incidence which resulted in a low baffle impedance. This dilemma led to the idea of utilizing resonances in attachments to the plate which could provide an effective impedance which exceeded that due to the added mass, over a suitable frequency band. By connecting these attachments through the plate only, the bending waves which appeared in solid plates were eliminated.

We embarked on a long theoretical and experimental program during which we used as resonators masses mounted on springs (studs), solid bars one-quarter wavelength long, T-bars, cylindrical sea chests, and combinations of these. We made measurements in the laboratory in air (comparing stiffened and unstiffened plates) using force generators, loud speakers and garden hoses as sources and accelerometers as sensors. We made measurements in water at USNUSRL comparing stiffened and unstiffened plates using hydrophones mounted flush with the plates as sensors. We have also excited these plates mechanically and compared the far field radiation in water as a function of frequency. We have tested single elements, a line array, and a 6 x 12 element array at Dodge Pond in transmission and reception, and we also have made measurements on the resonantly stiffened baffle of this array in air for comparison. We have attached thousands of resonators to the hull of a destroyer and measured the magnitude and bandwidth of the suppression of acceleration with the ship underway at sea. We have measurements on plates, stiffened by the same resonators, excited by acoustic waves

in air at various angles of incidence. We have extensive measurements in a sound booth in air on sections, at full scale, of resonantly stiffened 2 x 2 foot plates; we have 1:6 scale measurements on resonantly stiffened plates; and we have both 1:6 and 1:4 scale measurements on "aluminum block" baffles.

On the theoretical side we have done extensive work in calculating the effects of finite baffle impedance on the directivity of elements mounted in the baffle and on the radiation impedance of such elements. In the past we have assumed that the baffle could be described as having a uniform impedance over its surface. We recently published a study* which shows, as a function of plate thickness and frequency, the spacing at which discretely placed resonators are indistinguishable in effect from continuously distributed resonators.

A question has been raised about the relevance of measurements made in air to the behavior of the baffle in water. There are two effects to be considered: the difference in the loading of water and air on the baffle, and the difference in wavelength at a given frequency. In the frequency range of interest the impedance of the baffle must be 20 or 30 times that of water for the baffle to be useful (see below); whether the impedance of the medium is 1/20 that of the baffle, or zero, will make relatively little difference in the response of the baffle. The matter of the difference in wavelength is more subtle.

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* See Section III for a more detailed description of the study.

It is well known that if the wavelength in the medium exceeds the bending wavelength in a plate flexural waves will not be excited in the plate, and it is natural to suppose that waves observed in air might not be excited in water because the wavelength is approximately five times greater in water. We are not however dealing with unsupported plates, but with plates which are statically supported by bulkheads and dynamically supported by the attachments designed to increase the baffle impedance. The distinction is apparent from consideration of a simple bar. If unsupported it will respond simply as $1/\omega m$ to a uniform pressure; but if supported at one end, or at both ends, or anywhere else, it will respond resonantly as a cantilever, simple beam, etc. The work* by V. Mangulis and D. Yarmush on an infinite plate with parallel stiffeners shows this clearly; the pressure on the plate goes to zero at normal incidence when the plate between the stiffeners resonates as a clamped-clamped beam. In the absence of the stiffeners there would be no resonance at any frequency.

The experimental work is consistent with these observations. All the resonances seen at grazing angles are in evidence at normal incidence. In fact, we almost always get a greater increase in impedance over a wider frequency band with excitation at grazing angles than we do at normal incidence. We are not asserting that responses in air and water are identical, or that tests on small sections and large sections give the same results. We do assert that one can detect the presence of resonances in the frequency band of interest by tests in air, and that

* Discussed in Section III.

the magnitude of the average decrease in acceleration observed in air is a good measure of the increase in impedance achieved by resonant stiffening.

One significant question frequently raised about our work is, what confidence do we have in extrapolating measurements made on small baffles, or sections of the full array (e.g. Dodge Pond Test), to the full C/P array? We have two ways of estimating what will happen in a large array with finite baffle impedance. We know, as an upper limit on the effects of finite impedance, the loss at grazing angles due to a baffle of finite rigidity and infinite extent. We can superimpose on this the diffraction loss due to a finite baffle of infinite rigidity. The second means of estimating the loss due to finite baffle impedance is to determine the velocity distribution over an array in reception by matrix inversion, assuming various impedances looking into the elements. We have shown analytically that the effective baffle impedance is related to the assumed element impedance by the array factor. By solving for arrays of increasing length one can extrapolate to the full length (which we have done); or, if one can afford it, one can invert a matrix for the full array. Our calculations for the infinite array show that at 5^{o} from grazing one will get a loss of less than 0.5 db if the baffle impedance is 30 times that of water, and less than 3 db if the baffle impedance is 20 times that of water. This is the range of impedance that we have been achieving experimentally.

B. Mechanical Problems in the Construction of the Array Housing

We are aware of two mechanical problems related to our proposed sonar keel configuration which have been of concern to the Navy. One of these has to do with the strength of the structure and the other with the reliability of the proposed electron beam welding technique. We have not been authorized to work on either of these problems during the last year so that our position on structural questions is essentially that described in March 1966 in our "Recommendations for Installation of ESS of the USS SPOKANE," in the final section by I. Melnick. To summarize briefly, we found that a 20 foot radius of curvature, if uncorrected in the signal generator and beamformer, will cause a one-way loss of gain of 0.9 db at the upper end of the sonar band for an 8' high array in the worst case (array steered to broadside), where the loss is tolerable. The loss decreases to zero at end-fire. With this radius a plate thickness of less than an inch appears adequate. We have been directed to use a radius of 40 feet. We think this decision does not represent a good balance among considerations of system performance, complexity of electronics, structural integrity, and system cost. If this requirement stands we would propose to rearrange the sea chests by staggering alternate rows and moving the rows closer together. This will introduce the sea chests as effective members in resisting buckling of the shell. It will also increase the array factor. It will increase the system cost and weight and present problems in making watertight subdivisions of the keel.

The uncertainty at this time about the reliability of electron beam welding is very unfortunate because the matter could have been settled by a modest investment in a systematic test program. As it is we have only shoddy and conflicting evidence to go on, and the opinion of manufacturers of the welding equipment. We were given some free samples of electron beam weldments which were sent to TMB for examination. These welds proved to be unacceptable. We had some other samples tested for yield, ultimate and fatigue strength, and these proved as strong as the weaker material (316 stainless). We had some sea chests welded to a plate. This specimen passed the full shock series at the UERD facility of TMB. And we are assured by the equipment manufacturers that good welds can be consistently made. There is certainly no real evidence to the contrary. The cost of our proposed test program is $8000.*

C. The Protection of the Baffle from Cavitation Damage

In the TRG proposed configuration the individual transducer elements are supported in sea chests with the active face of the element flush with the baffle. This active face has a rubber boot attached to it. There are two reasons why one might want to bond a boot to the baffle as well, with circular cut-outs for the elements. One of these reasons is that there is a possibility that flow noise would be reduced

* TRG has retained, on company funds, Prof. Clyde M. Adams of MIT as a consultant on this problem. The results of his work will be sent to PTD as soon as available.

by the boot (this is not certain). The second reason is that the boot would protect the baffle from damage due to cavitation caused by the acoustic array. Against these advantages must be weighed the difficult problem of maintaining such a boot. Most of the initial installation could be made by vulcanizing the boot to the baffle in the factory, leaving strips exposed in the vicinity of joints to be welded. One would then hope to finish the job in the yard, but we are not confident that we can get a smooth surface and a strong air-free bond under shipyard conditions.

We suggest that it might be quite feasible to omit the boot. At least tests should be made both ways. We believe that the problem of cavitation damage might be solved by sensing the onset of cavitation at the electrical terminals of each element and automatically limiting the drive to that level. Once cavitation has started at a given element the acoustic power output from that element does not increase with input power (it usually decreases). The additional power goes into non-acoustic energy, part of which causes cavitation damage to structure. The situation with regard to predictability of the far field patterns and radiation impedances is probably improved as compared to fixing the relative drive levels and driving part of the array into hard cavitation. D. Carson of NEL has suggested that the drive level on the entire array can be limited to a value that produces an indication of cavitation on some predetermined number of elements. This will tend to limit the effective shading produced by the automatic control.

Another possibility would be to sense cavitation in the acoustic field using a number of tiny flush elements interspersed among the transducer elements.

If it is possible to eliminate the boot in the sparse array configuration this may be one of the significant advantages of the sparse over the dense array, since the latter may require a boot to achieve adequate surface smoothness.

D. Summary

We have made a large number of studies of the various factors affecting the conformal/planar array. We believe that the studies indicate that the array could be built now without any further major calculations and tests.

The array should be located in a sonar keel hull configuration.* The transducer elements should be placed within sea chests attached to the steel plate (about 3/4" thick) which forms the air backed array baffle. The sea chests themselves act as mechanical resonators which rigidize the baffle over the frequency band from 2000 cps to 3000 cps; additional resonators can be attached to the steel plate between sea chests. The transducer elements occupy about 31% of the total array area. The array consists of about 12 rows, spaced about 8" apart, and about 225 columns, spaced on average, 8" apart. The circular piston of an array element has a diameter of 5". The transducer element**

* See Section V for weight Summary.

** See Section IV.

weighs about 50 lbs.* In transmission one has to switch inductors so that the entire 2000 - 3000 cps band can be covered by three bands of about 300 cps bandwidth each with a different inductor in each band, but in reception the element operates over the entire 1000 cps bandwidth with a single inductor. The reasons for the choice of some of the array parameters and some additional comments on some factors influencing the C/P array design may be found in the TRG report "Recommendations for Installation of ESS on the USS SPOKANE" (March 1966).

It should be noted that our experimental tests show that the baffle will also be rigidized at frequencies below 2000 cps (see Section III.B); therefore the C/P array could be used for passive listening at lower frequencies. Actually at lower frequencies the baffle does not have to be as rigid as at the higher frequencies: the attenuation of the signal near end fire due to the finite impedance of the baffle depends on the length of the baffle traversed, where length is measured in wavelengths, and at a fixed point on the array the length (in wavelengths) traversed by the signal decreases as the frequency decreases.

*See Section V for weight breakdown.

II. RADIATION THEORY

In underwater acoustics one usually is concerned with three acoustic field quantities for any given sound radiation system; the quantities are:

1. the far field radiation pattern;
2. the radiation impedance of each element in the system;
3. the location and value of the maximum pressure in the acoustic field.

The reasons for being concerned with the first quantity are obvious; the directivity of the acoustic system affects both transmission and reception of sound.

The radiation impedance can affect the electrical phasing and relative voltages which have to be applied to individual elements in the system if one wants to obtain a given far field pattern in transmission; or it can determine the phases and voltages one obtains in reception, and thus affect the way in which such individual signals have to be added for optimum reception by the system. However, if one applies "Carson's cure"[1] to the array elements, the effects of the changes in radiation impedance can be made negligible. Thus the variation of radiation impedance with steering angle, frequency, or position

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[1] D.L. Carson, Diagnosis and Cure of Erratic Velocity Distributions in Sonar Projector Arrays, J. Acoust. Soc. Am. 34, 1191 (1962). Velocity Control is achieved by making the transducer impedance much larger than the radiation impedance.

in the array is not an important factor in the array design if one considers only the effects of the radiation impedance on the inputs or outputs at the electrical terminals. However, since the radiation impedance is proportional to the average pressure on the element, one can sometimes use the maximum radiation impedance as a measure of the maximum pressure in the near field. The third acoustic field quantity listed above puts an upper limit on the power which the acoustic system can radiate because the maximum pressure in the acoustic field should not exceed the local pressure in the absence of the acoustic signal, otherwise cavitation will set in, and the signal will deteriorate. We have investigated the near field pressure for large arrays[2,3] and infinite arrays[4,5] and have found that the maximum is frequently situated on the pistons in the array; in that case since

$$p_{av} A = Z_R v \tag{1}$$

where p_{av} is the average pressure on the piston, A is the piston

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[2] V. Mangulis, S. Gardner, A. Novick, W. Graham, A Study of Conformal Arrays for Long Range Sonars, Phase II, TRG, Inc., Report No. TRG-142-TR-2(1963), vol. II, AD 336256, Appendix IV (Confidential).

[3] V. Mangulis, Near-field Pressure for a Steered Array of Strips, J. Acoust. Soc. Am. **38**, 78 (1965).

[4] V. Mangulis, Near-Field Pressure for an Infinite Array of Strips, IEEE Trans. Sonics & Ultrasonics **SU-13**, 49 (1966).

[5] V. Mangulis, Near Field Pressure for an Infinite Phased Array of Circular Pistons, TRG, Inc., report No. TRG-023-TN-66-18 (1966); to be published in J. Acoust. Soc. Am., February 1967 issue.

area, Z_R is the radiation impedance, and v is the piston velocity, one can find $|p_{av}|$ for each piston from the radiation impedance and use maximum $|p_{av}|$ as an approximation to the maximum pressure in the near field; if $|v|$ and A are the same for all pistons in the array, $|p_{av}|$ will be maximum when $|Z_R|$ is maximum.

Thus in a practical design study one can tolerate large inaccuracies in the calculation of radiation impedance, if one uses "Carson's cure" in the transducer design, and if one calculates the actual peak pressure. Alternately, one can neglect the calculation of the near field peak pressure, if one is willing to use the average pressure instead.

To calculate the acoustic field quantities one usually assumes a mathematically tractable model for the sonar array, for example, an array situated on an infinite rigid planar baffle or on an infinite rigid baffle in the shape of a circular cylinder. Of course, no practical baffle is infinite, and most baffles are not rigid. The C/P array may be situated underneath the ship, and the array housing and the ship's hull will form a corner which reflects sound and therefore the corner has to be taken into account in the calculations.

The most important effects are discussed below. Diffraction effects due to the curvature and the finite extent of the baffle are described in subsection II.A. Transient effects due to the switch-on and switch-off of the array, and due to the multi-frequency operation of the array, are described in subsection II.B. Effects associated with the flexibility of the baffle are discussed in section III.

If diffraction and nonrigidity of the baffle can be neglected (as shown below), then one can use the existing radiation theory and computer programs to compute the far field patterns, the radiation impedances, and the near field pressure as if the array were situated on an infinite rigid plane baffle.[6] An approximate expression for the peak near field pressure is also available.[5]

A. Diffraction Effects

1. Reflection at the hull of the ship

An extensive study of the effects of a rigid infinite corner on the far field pattern, radiation impedance, and near field pressure of a piston has been performed.[7,8,9] The corner simulates the intersection of the sonar housing with the hull of the ship. A summary of the conclusions is presented below.

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[6] V. Mangulis, Baffle Reflections, Diffraction, and Nonrigidity in the CONTACT Sonar System, TRG, Inc., report No. TRG-011-Memo-64-1 (Confidential); (1964).

[7] V. Mangulis, Pistons on Corners, TRG, Inc., report No. TRG-142-TN-63-3 (August 1963), also published in J. Franklin Institute, Vol. 279, pp. 124-135 and pp. 200-208 (1965).

[8] A. Kane, T. DeFilippis, V. Mangulis, Effect of Reflections from the Ship's Hull on the CONTACT Sonar Array (U), TRG, Inc., report No. TRG-023-Memo-65-2 (1965); (Confidential).

[9] V. Mangulis, On the Radiation Impedance of a Piston on a Corner, TRG, Inc., report No. TRG-023-TN-65-2 (1965).

Let us assume that the piston is in an infinite rigid vertical half plane; the other half plane forming the corner can vary from an almost horizontal to a vertical corner angle ψ (see Fig. 1). Then if the piston is at least a wavelength away from the edge of the corner, the pressure field can be approximated by the superposition of two pressure fields from pistons on rigid infinite planar baffles without corners, placed at an angle with respect to each other; the two fields are due to the original piston and its image (in phase with the actual piston) as shown in Fig. 1, each radiating in a half space. In general there is a region in space in which only the pressure field from the actual piston exists, and there is another region in which the fields from the actual piston and the image piston are superimposed; the boundary between the two regions is at the angle $2\psi - \pi$, and the approximation does not hold very well at this boundary or at the edge of the corner.

Let us now consider the far field radiation patterns for a C/P array. The array will be placed at a 20° tilt angle with respect to the vertical, see Fig. 2. Let us disregard the diffraction effects due to the finite size or curvature of the sonar keel and the ship, then we can approximate the situation by an array on an infinite corner (assuming the actual array, if it is curved, is phased properly to a plane, etc.), as shown in Fig. 2. Let the mean plane through the ship's hull in the vicinity of the sonar keel make an angle α with the horizontal, then the corner angle $\psi = 90^{\circ} + 20^{\circ} + \alpha$. Let the

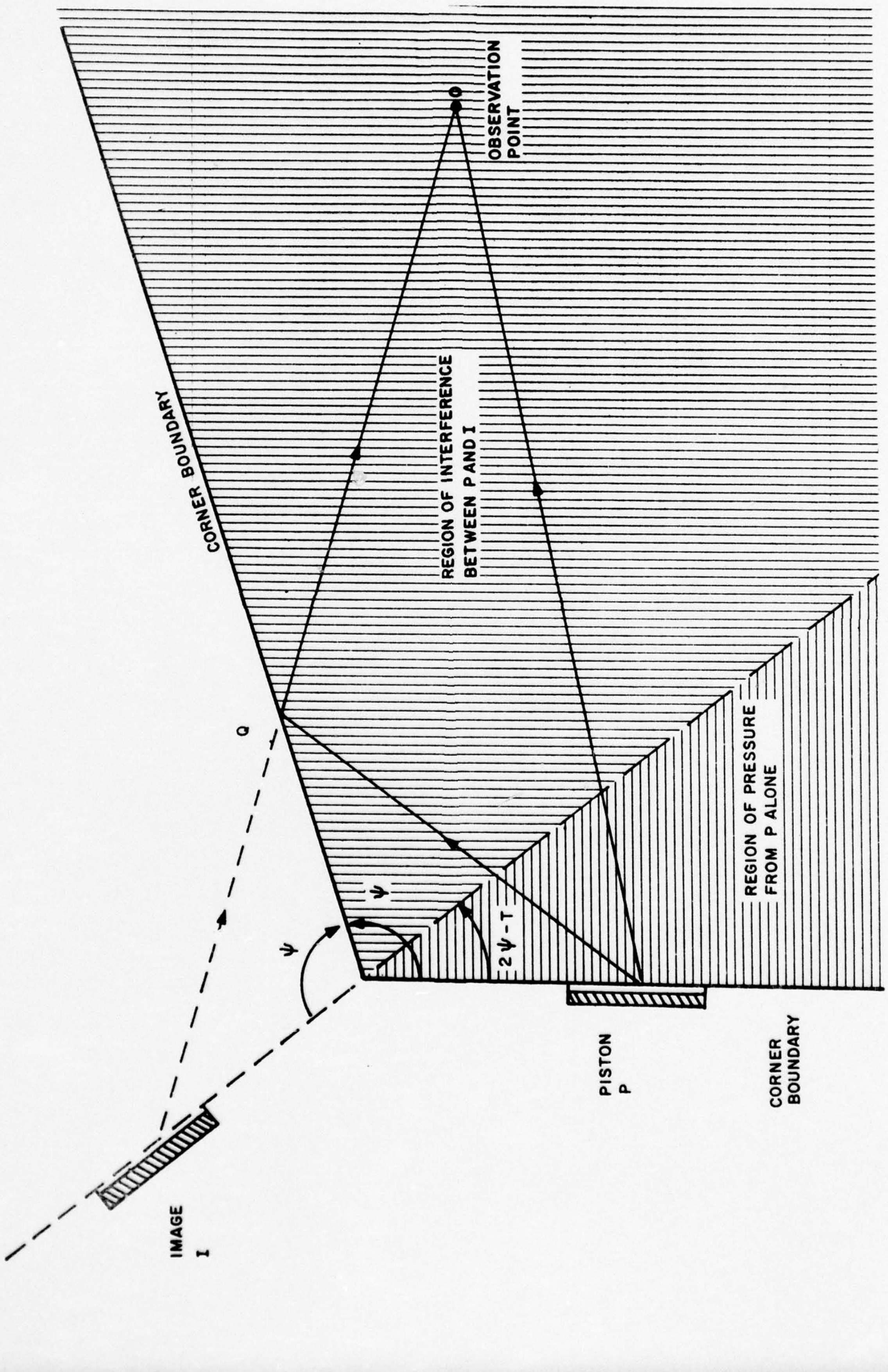

FIGURE II.A-1 IMAGE FORMED BY THE PRESENCE OF CORNER

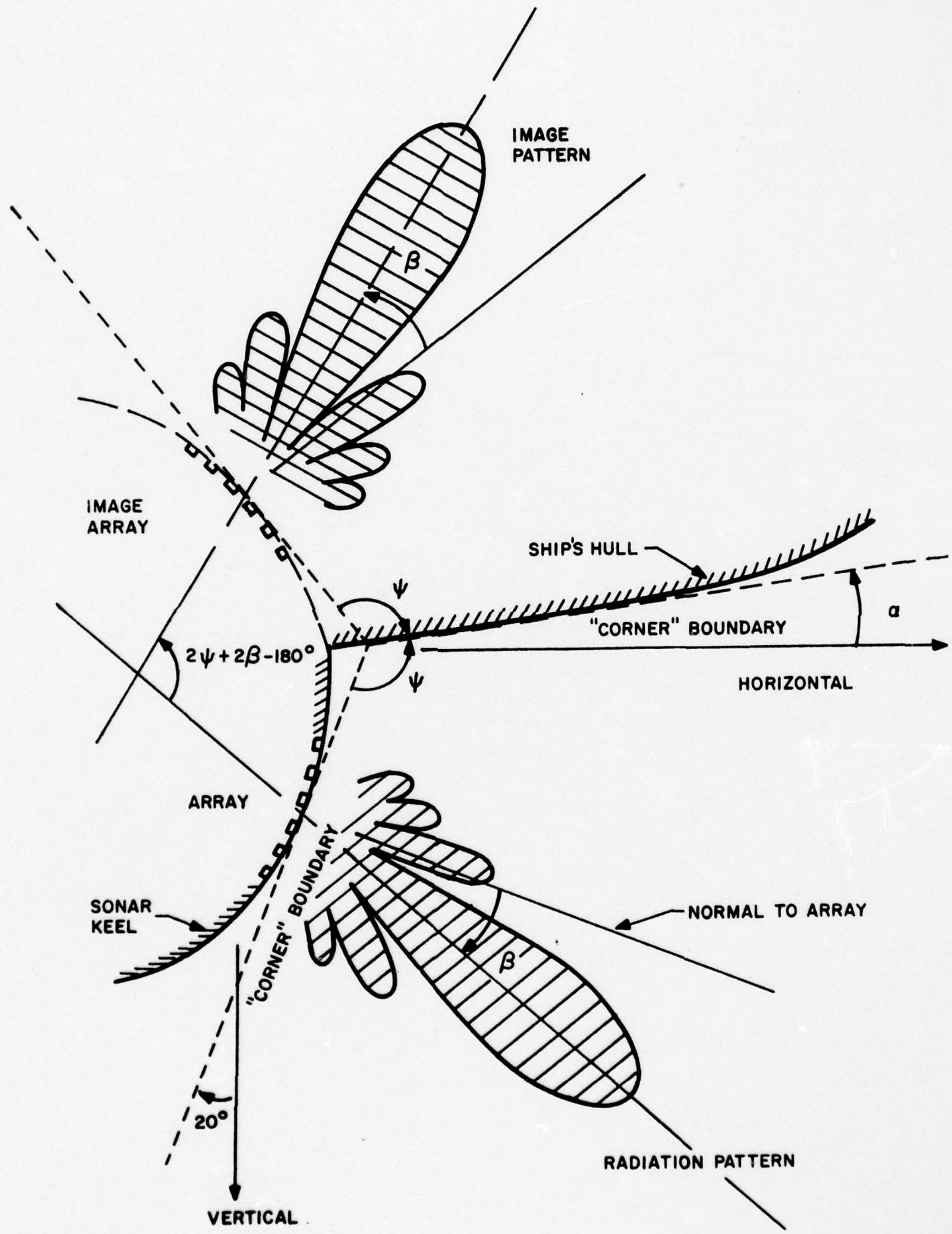

FIGURE II. A.-2 ARRAY ON THE SHIP

far field radiation pattern be steered down by an angle β with respect to the normal of the array, then the angle between the directions of the main beams for the actual and image arrays is $2\psi + 2\beta - 180^{\circ}$ $= 2(\alpha + \beta + 20^{\circ})$. The total pattern is the superposition of the two patterns. Thus interference between the actual and image arrays will be most troublesome when both main beams are pointing in the same direction. Such a situation could develop if $\psi = 90^{\circ}$ and $\beta = 0^{\circ}$; however, in such a case the point of reflection Q in Figure 1 will be halfway between the array and the far field observation point O; i.e., if the target is in the horizontal plane at 2 kiloyards, the ship's hull would have to extend to 1 kiloyard to produce the image main beam.

When $\beta = 0$ and $90^{\circ} \leq \psi \leq 135^{\circ}$ the distance s between the edge of the corner and the point of reflection Q is given by

$$s = d \cos 2\psi / \cos \psi$$

where d is the distance between the edge of the corner and the source.

Thus when the ship's hull is nearly horizontal it is not wide enough to produce reflections, while if the hull is at some angle above horizontal the image array points in a different direction and interference is produced by sidelobes only. In either case the effect of the image array can be neglected, and the far field patterns can be calculated as if the array were located on an infinite planar baffle.

The above arguments are supported by experimental patterns measured at U.S. Navy Underwater Sound Laboratory in 1964. Figures 3 and 4 show patterns in the vertical plane for an array consisting of 6 rows of

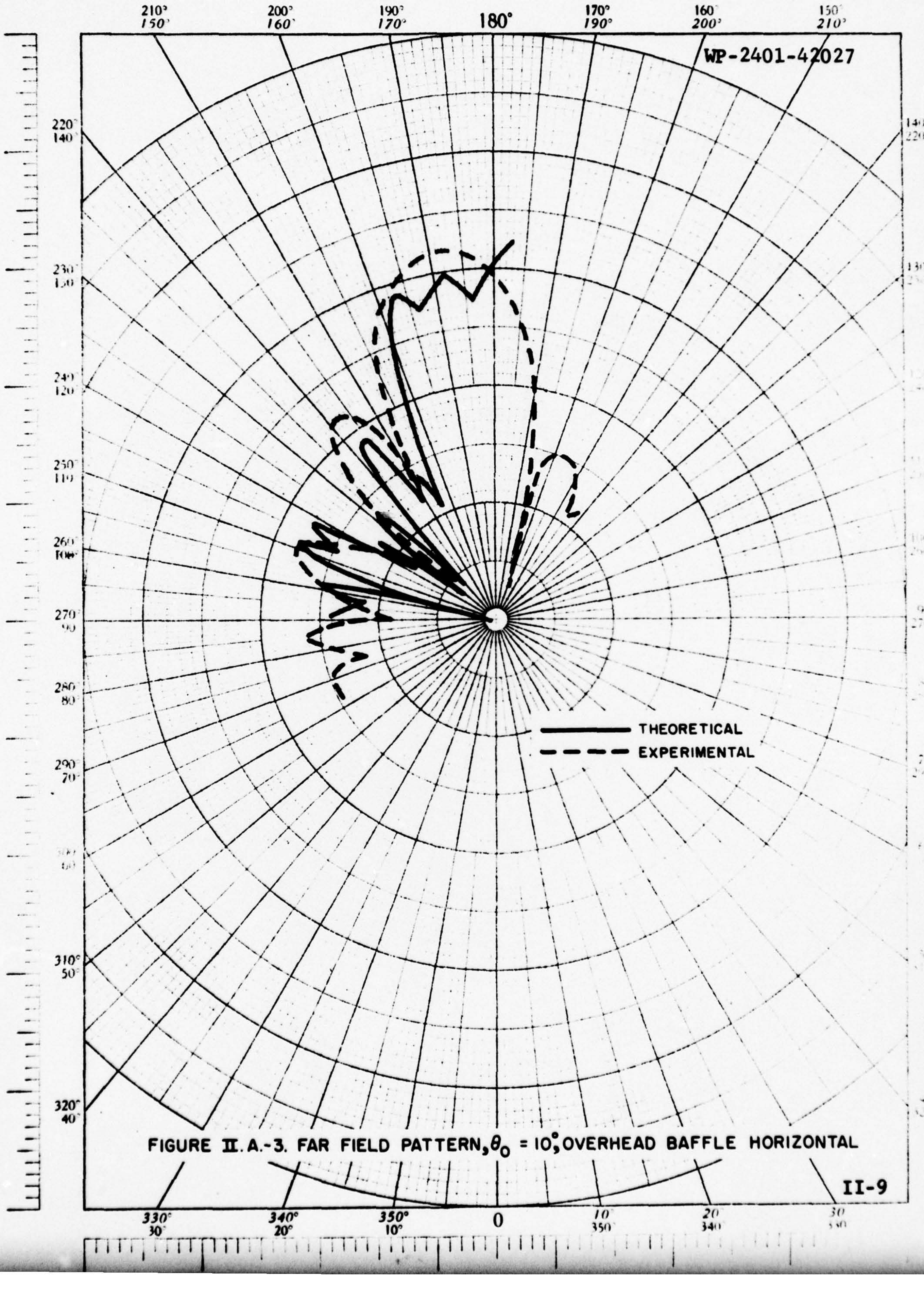

FIGURE II.A.-3. FAR FIELD PATTERN, θ_0 = 10°, OVERHEAD BAFFLE HORIZONTAL

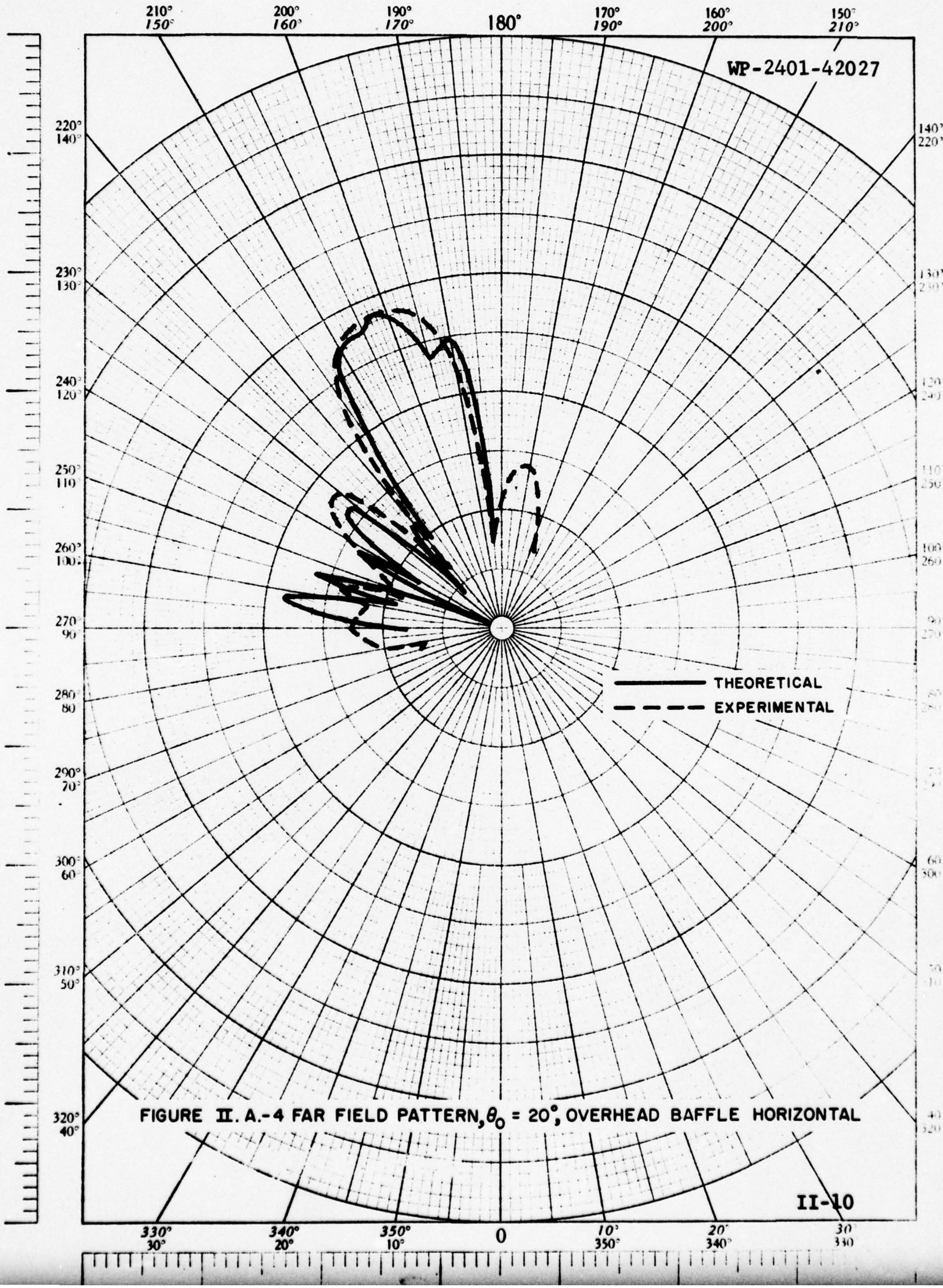

FIGURE II. A.-4 FAR FIELD PATTERN, θ_0 = 20°, OVERHEAD BAFFLE HORIZONTAL

12 elements each; a movable overhead baffle was hinged to the sonar keel to simulate the ship's hull. The overhead baffle was horizontal, the array was steered to 10^{o} and 20^{o} down from the horizontal plane in a vertical plane normal to the array. The radiation theory including the effect of an infinite corner predicts interference ripples in the shape of the main beam (solid curves in Figures 3 and 4), but such ripples were not observed experimentally because the simulated ship's hull was too short to produce the interference.

According to the above approximation for $\psi > 90^{o}$ the image field does not exist on the face of the actual piston (see Fig. 1), therefore for $\psi > 90^{o}$ the near field peak pressure and the radiation impedance should be approximately the same as for the same piston in an infinite planar baffle. This is confirmed by some exact calculations:[7] for $\psi = 135^{o}$ the radiation self-resistance of a square piston of sides .22 wavelengths long, a wavelength from the edge of the corner, is 0.04% lower than for the same piston in an infinite planar baffle, while the mutual radiation resistance coefficient for two square pistons of the same size, about a wavelength apart, differs by 5.3% in magnitude from the coefficient for pistons in a planar baffle. Such exact calculations are not available for the radiation reactances, but since the radiation resistances are unaffected by the presence of the corner, and the radiation reactances are related to the resistances by the Kramers-Kronig or dispersion relations (or vice versa), one can show that the reactances will also be independent of the corner angle ψ.[9]

As explained above, the mean plane through the array is tilted with respect to the vertical plane, and the ship's hull is at least a few degrees above horizontal, therefore ψ for the corner formed by the sonar keel and the ship's hull always exceeds 90°. A physical explanation why the effect of the image array on the actual array is negligible is as follows:

1) Since ψ exceeds 90°, the actual and the image arrays are on two different planes with an edge between them (for $\psi = 90^{\circ}$ those planes would coincide to form a single infinite plane); this edge shields the radiation coming from the image array so that the radiation is confined approximately to the shaded interference region in Figure 1, and thus does not reach the actual array;

2) Since the ship's hull is at least a few degrees above horizontal, the main beams from the actual and the image arrays are always pointing in different directions, and even in the near field the energy flow is primarily concentrated in the direction in which the array is steered;[10] thus very little energy from the image array would flow in the direction of the actual array.

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[10] See collection of papers on the near field in the July 1964 issue of U.S. Navy J. Underwater Acoustics (Confidential).

As an overestimate of what the image array would do to the radiation impedances of the array we can eliminate 1) above by making $\psi = 90^{\circ}$, then the actual and image arrays are located on the same infinite plane with no edge in between. Tables II.A.I-III show the maximum, minimum, and average radiation impedances $\hat{Z}_R/\rho c\pi b^2 = Z_R = R_R + iX_R$, where b is the piston radius, ρ the density of water, c the sound velocity, for an array of 9 rows and 228 columns operating at 3000 cps, spacing between rows 0.197 meters, spacing between columns 0.181 meters, piston radius 0.0635 meters, for several steering angles θ_o and ϕ_o, see Figure 5. The numbers in parentheses after the impedance values denote row and column numbers. The edge of the corner is 0.406 meters above first row.

The numbers presented in Table I for $\theta_o = 0^{\circ}$ are really an overestimate of an overestimate since then both 1) and 2) above are eliminated, i.e., we have eliminated both the shielding effect of the edge and also have permitted the actual array and the image array to radiate in the same direction. Table I has been included for completeness to show that even in such a case the radiation impedances do not become outlandish; but the impedances which one will encounter in the actual system are given in Tables II and III. Thus the overestimate shows that the effect of the corner on the radiation impedances will be rather small, as already concluded in Reference 6.

Table II. A-I 9 x 228 array, $\theta_o = 0°$

			ϕ_o=0°(endfire)		ϕ_o = 30°		ϕ_o=90°(broadside)	
Infinite plane baffle	Radiation impedances	Max R_R	1.795	(5,1)	0.948	(3,220)	0.464	(4,4)
		Min R_R	0.221	(1,228)	0.239	(5,209)	0.227	(5,226)
		Max X_R	4.172	(5,2)	0.603	(1,225)	0.421	(1,1)
		Min X_R	0.412	(2,228)	0.022	(5,215)	0.078	(8,2)
		Average Z_R	1.412 + i2.733		0.640 + i0.295		0.364 + i0.227	
		Cavit.lim power out, watts	0.141×10^5		1.250×10^5		2.247×10^5	
Right-angled corner baffle	Radiation impedances	Max R_R	2.884	(1,1)	1.028	(2,201)	0.464	(2,2)
		Min R_R	0.209	(1,228)	0.178	(5,209)	0.230	(3,226)
		Max X_R	4.074	(3,2)	0.616	(1,224)	0.446	(9,1)
		Min X_R	0.406	(8,228)	-0.011	(4,215)	0.045	(8,2)
		Average Z_R	1.709 + i2.592		0.647 + i0.287		0.359 + i0.232	
		Cavit. lim. power out, watts	0.145×10^5		1.133×10^5		2.092×10^5	

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Table II.A-II. 9 x 228 array, $\Theta_o = 20°$

			ϕ_o=0°(endfire)		ϕ_o = 30°		ϕ_o=90°(broadside)	
Infinite plane baffle	Radiation impedances	Max R_R	1.580	(8,1)	0.968	(5,219)	0.529	(9,2)
		Min R_R	0.202	(1,228)	0.217	(1,228)	0.184	(8,3)
		Max X_R	1.869	(9,196)	0.690	(9,225)	0.441	(9,1)
		Min X_R	0.411	(8,228)	-0.091	(8,214)	0.082	(3,2)
		Average Z_R	0.952 + i1.159		0.676 + i0.304		0.378 + i0.234	
	Cavit. lim power out, watts		0.347×10^6		1.142×10^5		1.963×10^5	
Right-angled corner baffle	Radiation impedances	Max R_R	1.471	(8,1)	1.025	(3,1)	0.519	(3,2)
		Min R_R	0.193	(1,228)	0.208	(1,228)	0.166	(8,3)
		Max X_R	1.879	(9,1)	0.693	(9,225)	0.456	(9,1)
		Min X_R	0.408	(8,228)	-0.083	(8,214)	0.062	(3,5)
		Average Z_R	0.839 + i1.166		0.688 + i0.298		0.374 + i0.238	
	Cavit. lim.power out, watts		0.321×10^5		1.022×10^5		1.918×10^5	

Table II. A-III.9 x 228 array, $\Theta_o = 45°$

			ϕ_o=0°(endfire)		ϕ_o = 30°		ϕ_o=90°(broadside)	
Infinite plane baffle	Radiation impedances	Max R_R	1.127	(9,3)	1.257	(9,218)	0.679	(4,2)
		Min R_R	0.188	(1,228)	0.196	(1,228)	0.231	(8,4)
		Max X_R	1.405	(9,1)	0.833	(4,1)	0.536	(2,1)
		Min X_R	0.417	(1,213)	0.406	(7,228)	-0.006	(7,3)
		Average Z_R	0.665 + i0.893		0.761 + i0.597		0.461 + i0.284	
	Cavit. lim. power out, watts		0.412×10^5		0.734×10^5		1.488×10^5	
Right-angled corner baffle	Radiation impedances	Max R_R	1.134	(9,3)	1.244	(9,1)	0.670	(4,2)
		Min R_R	0.181	(1,228)	0.188	(1,228)	0.221	(1,226)
		Max X_R	1.408	(9,1)	0.786	(4,1)	0.530	(2,1)
		Min X_R	0.413	(1,213)	0.406	(7,228)	-0.005	(7,3)
		Average Z_R	0.681 + i0.896		0.787 + i0.555		0.458 + i0.285	
	Cavit. lim.power out, watts		0.418×10^5		0.794×10^5		1.530×10^5	

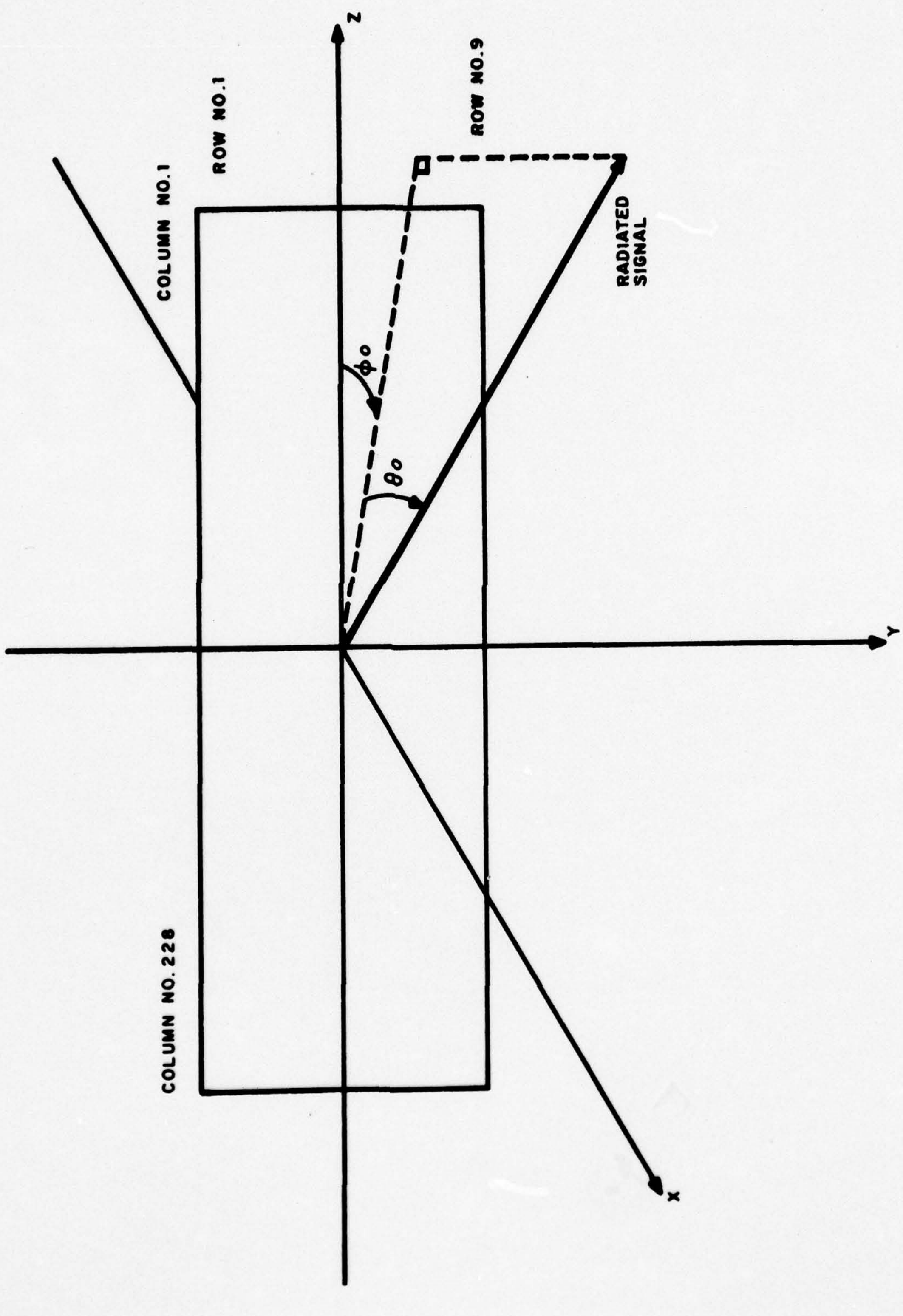

FIG. II A.-5. GEOMETRY OF THE ARRAY

Tables I - III also show the cavitation limited power output, where the cavitation limit was determined by setting the largest average pressure on a piston equal to 1.5 atmospheres. The power output is also unaffected by the presence of the corner.

To summarize: for the corner angles which will be encountered by a C/P array the effects of reflections at the ship's hull on radiation impedance, maximum near field pressure, and far field radiation patterns will be negligible.

2. Curvature of the baffle

We have studied the acoustic effects of curved baffles because such baffles may be used for structural reasons.

Consider the acoustic radiation from a piston on an infinite rigid circular cylinder. If the radius of the cylinder exceeds a few wavelengths, then (see Fig. 6) in front of the piston the field is roughly the same as for the same piston on an infinite planar baffle; in back of the cylinder the field can be considered to be equal to zero; those two regions are divided by the "penumbra region" in which a gradual transition in field strength takes place, and the width of this transitional region is $\theta \approx (ka)^{-1/3}$, see Fig. 6, where $k = 2\pi/\lambda$, λ is the wavelength, and a the radius of the cylinder.[11,12,13] If the radius

[11] V.A. Fock, The Surface Current Distribution Induced by an Incident Plane Wave, Zh. Eksper. Theor. Fiziki 15, 693 (1945).

[12] V.A. Fock, New Methods in Diffraction Theory, Phil. Mag. 39, 149(1948).

[13] M.L. Levin, On the Effect of Surface Curvature of a Convex Metallic Body on the Radiation of a Source on this Surface, Zh. Tekh. Fiz. 25, 2395 (1955).

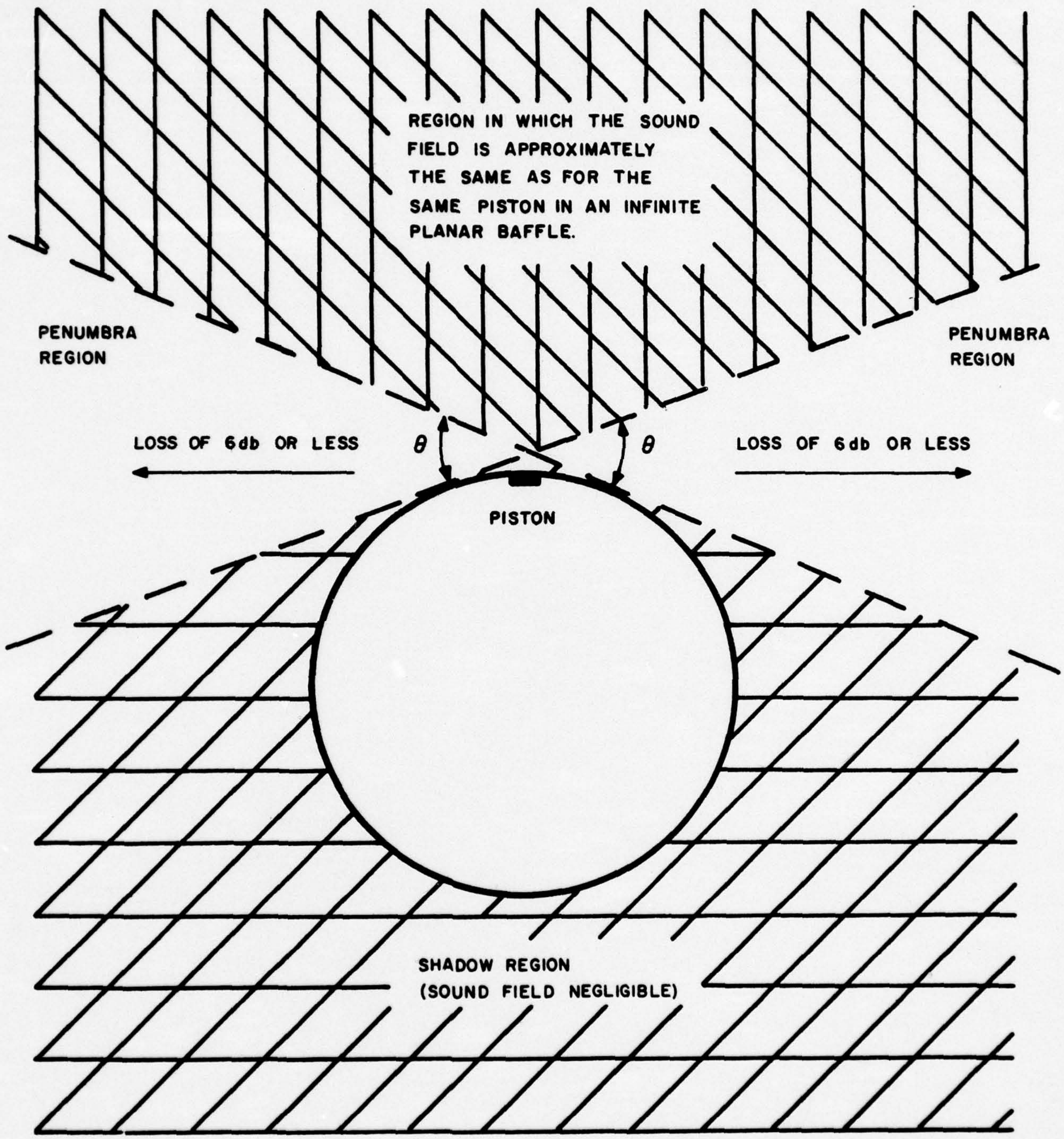

FIGURE II.A-6. DIFFRACTION BY A LARGE CIRCULAR CYLINDER

of curvature of the baffle in the vicinity of the array is $a \gtrsim 8\lambda$, then $\theta \gtrsim 15.5^{o}$. Only the local curvature matters; in a sonar keel the curvature changes at the bottom of the keel, but that part of the keel is a few wavelengths away from the array, and therefore will not affect the radiation pattern.

For an array on a sonar keel the penumbra region occupies directions in which we do not transmit or receive, and thus is irrevelant (see Figure 7). Thus we can expect that the radiation patterns of arrays on a sonar keel will closely resemble the patterns of arrays on an infinite rigid plane baffle, and the available computations confirm this conclusion.[14,15,16] Consider an array on an infinite cylinder, see Fig. 8. This array is driven with velocities of equal magnitude, and the phases are such that all pressures add in phase in the steering direction. Because of the effects of diffraction on the phases of the pressures of each piston, the required phase distribution is not exactly phased to a plane.[15] For a given steering angle, however, the difference

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[14] D.T. Laird and H. Cohen, Directionality Patterns for Acoustic Radiation from a Source on a Rigid Cylinder, J. Acoust. Soc. Am. 24, 46 (1952).

[15] V. Mangulis, A. Kane, E. Paige, Comparison of the Far-Field Patterns of an Array on a Cylinder with an Array on a Plane, TRG, Inc., report No. 023-TM-65-4 (1965).

[16] J. Gobins, Supplement to Report No. 023-TM-65-4, TRG, Inc., report No. 023-TM-66-31 (1966).

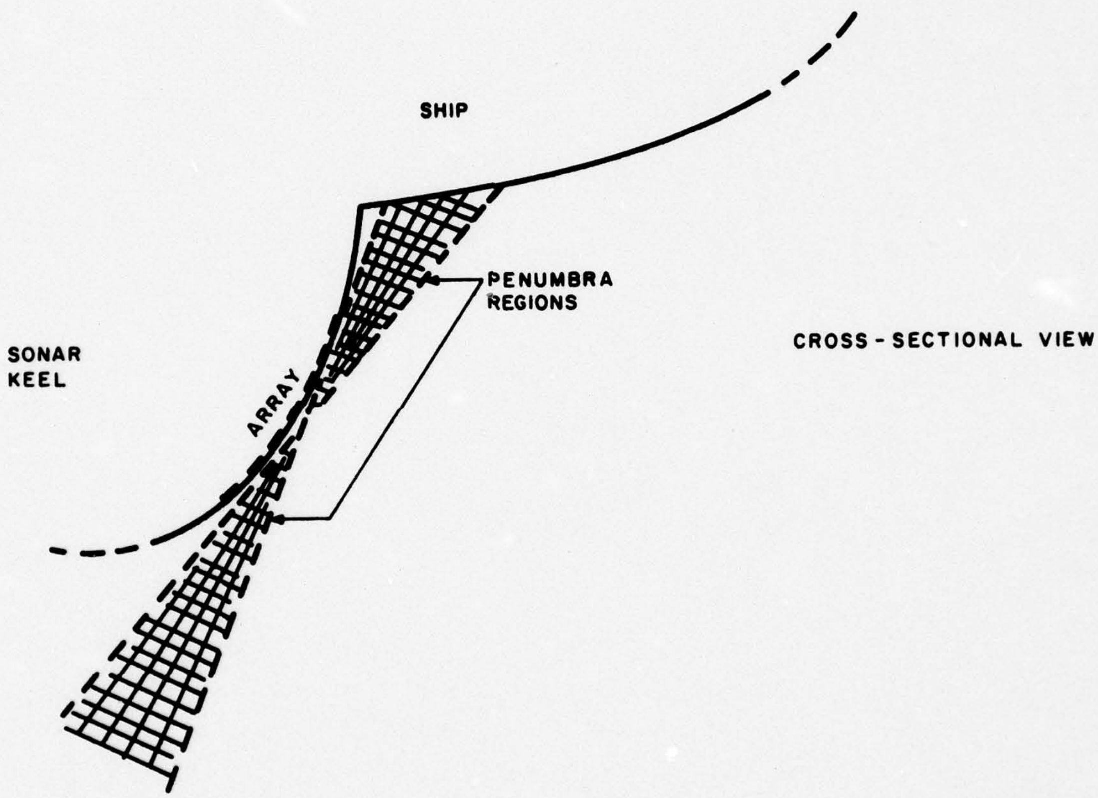
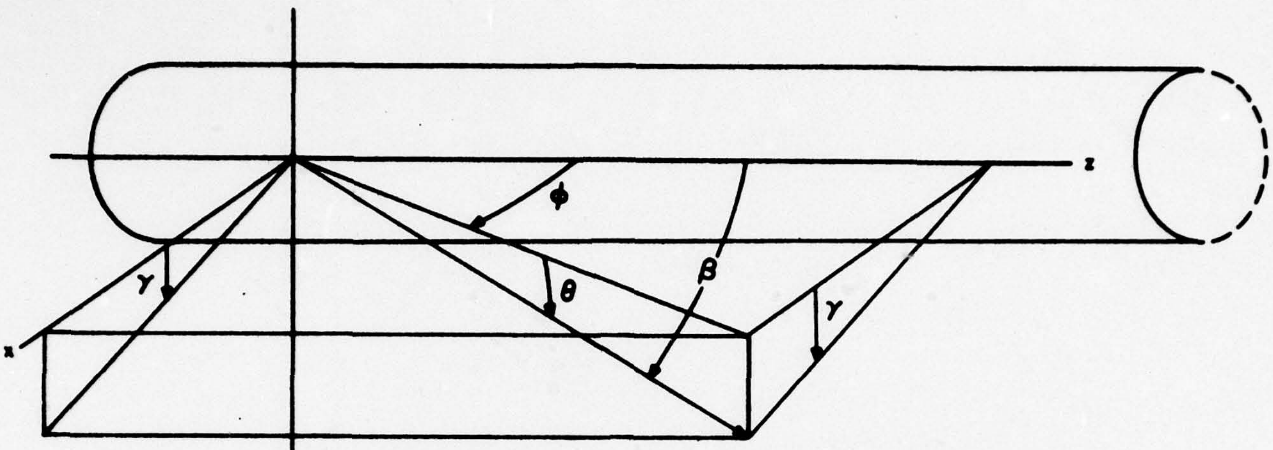
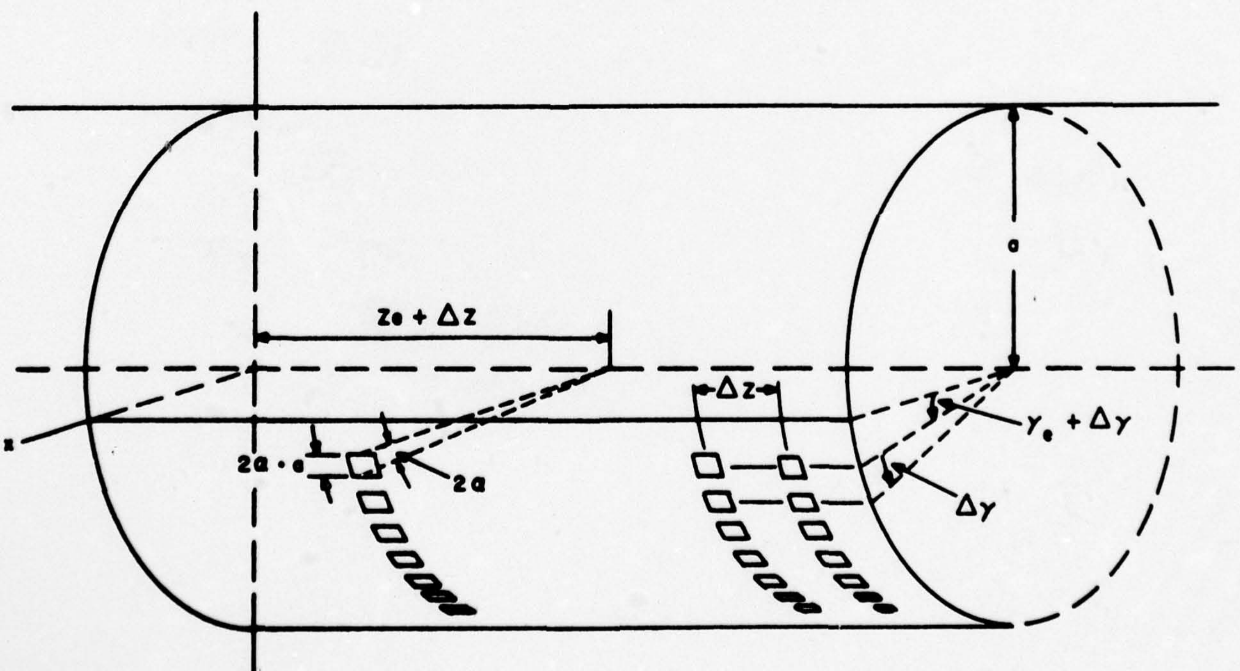


FIG. II.A.-7. DIFFRACTION BY SONAR KEEL



θ_0 AND ϕ_0 ARE THE DIRECTIONS TO WHICH THE BEAM IS STEERED IN THE θ AND ϕ PLANE.

8a. ANGLES



8b. DIMENSIONS

FIGURE II.A.-8. ANGLES AND DIMENSIONS FOR THE ARRAY ON A CYLINDER

between phases based on geometric path alone, and those necessary to make all pressures add in phase in the steering direction is nearly constant; deviations are of the order of one degree for the whole array.[15] For the parameters listed in Table IV we obtain[16] the far field radiation patterns vs. ϕ and vs. θ shown in Figs. 9 and 10. The parameter T represents the depression angle of the normal to the array on the cylinder, and is the tilt angle of the array on the infinite plane. The main beam is steered to end-fire in the azimuthal direction, but at a depression angle of 26.6°. The patterns for the arrays on a plane and on a cylinder differ by 1 db or less.

Let us now consider the near field and radiation impedances. We can infer that there should not be a significant change in radiation impedances because of diffraction due to curvature of baffles because we have shown that if baffles are roughly planar within a few wavelengths of the piston (i.e., the curvature is sufficiently large so that it appears to be approximately planar within this region) then the piston radiates a field which is almost the same as with an infinite planar baffle, except for some diffusion of the field into the shadow regions. Since the energy which diffuses into a shadow region comes mainly from the adjacent insonified region, and since the shadow regions are at least a few wavelengths from the piston, the field at the piston remains relatively undisturbed. This argument is confirmed by Greenspon's and Sherman's[17] calculations for pistons on cylinders and spheres.

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17 J.E. Greenspon and C.H. Sherman, Mutual-Radiation Impedances and Near-Field Pressure for Pistons on a Cylinder, J. Acoust. Soc. Am. **36**, 149 (1964).

TABLE II. A-IV

$f = 3500$ cps.

$c = 1534.9$ meters/sec.

$Z_o = 0.0563$ meters

$N = 240$

$M = 7$

$\Delta_z = 0.1777$ meters

$a = 2.400$ meters

$\alpha = .02338$ radians

$\gamma_e = 2.0°$

$\Delta\gamma = 5.0°$

$T = 22°$

$\theta_o = 26.6°$

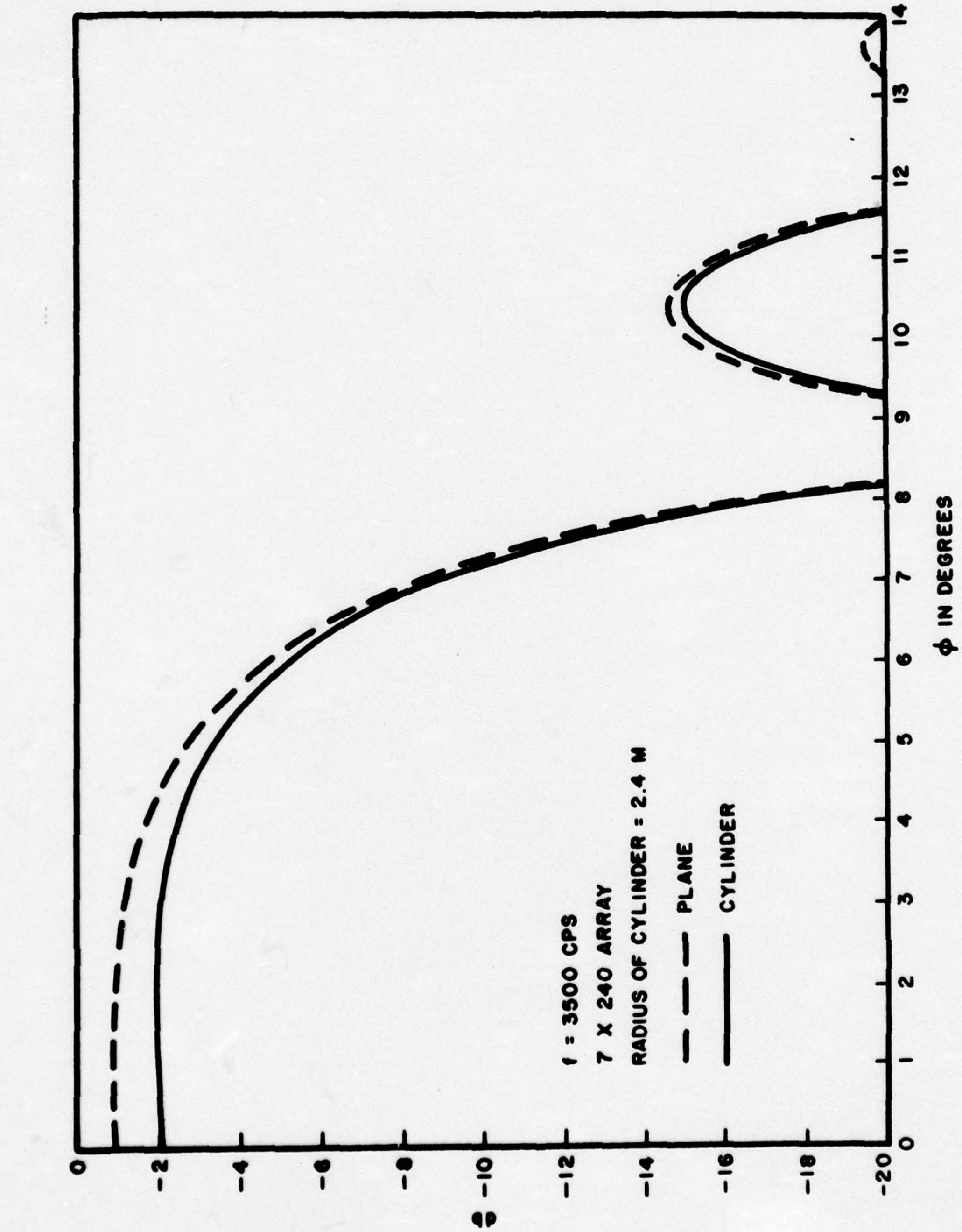

FIGURE II.A.-9 FAR FIELD PATTERN VS. ϕ FOR AN ARRAY ON AN INFINITE PLANE AND CYLINDER, WHEN $\theta = \theta_0 = 26.6°$ AND $\phi_0 = 0°$.

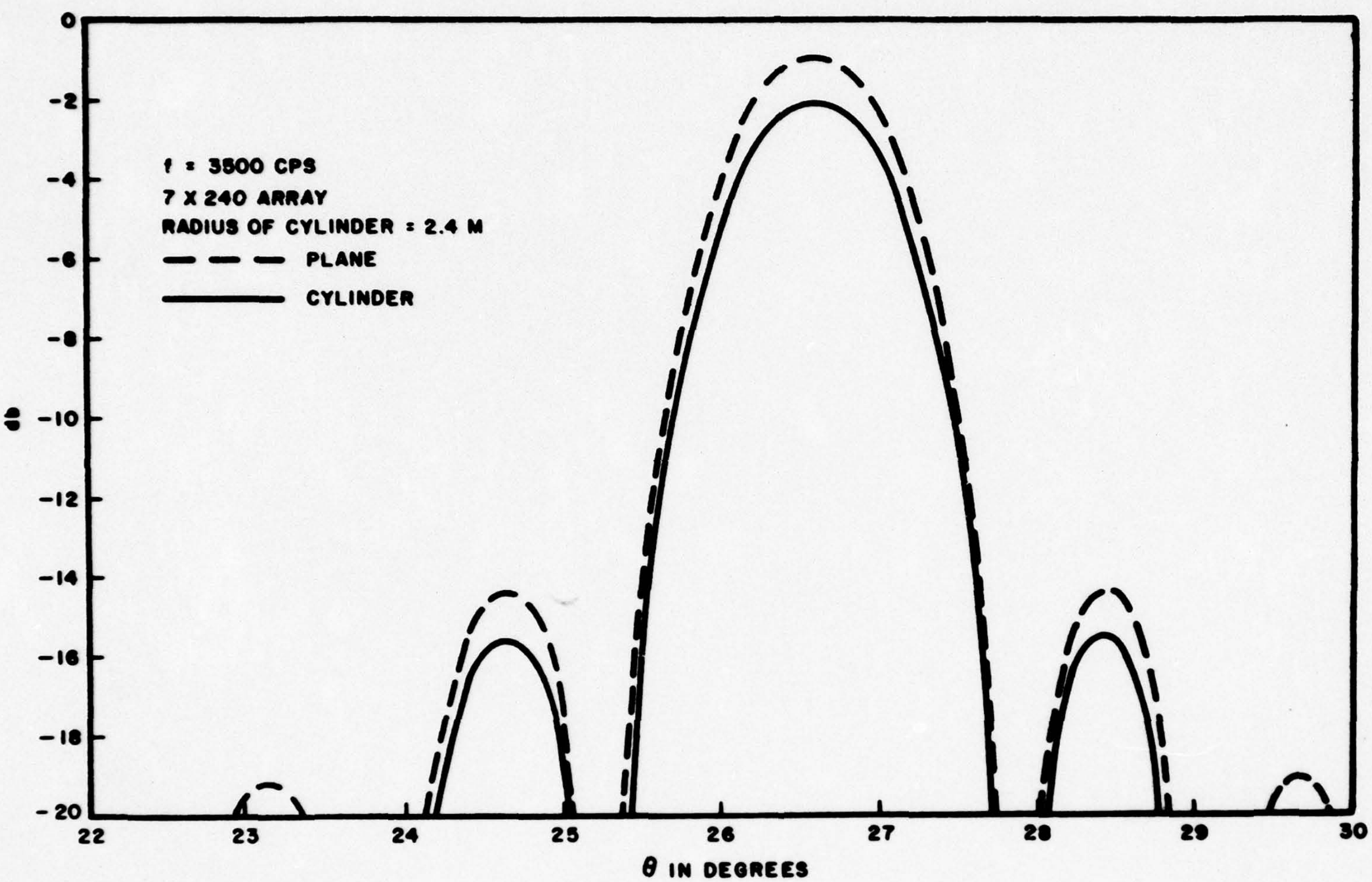

FIGURE II.A.-10 FAR FIELD PATTERN VS. θ FOR AN ARRAY ON AN INFINITE PLANE AND CYLINDER, WHEN $\phi = \phi_0 = 0°$ AND $\theta_0 = 26.6°$

3. Finite extent of the baffle

When we considered the effects of curvature in the previous subsection, the cylinder was assumed to be infinite. However, the actual proposed sonar keel is finite, and therefore one should also investigate what effect the sudden termination of the baffle will have on the radiation from the array. Since we showed in the previous subsection that the curvature effects are negligible, we might as well consider a plane baffle (instead of a curved baffle) which suddenly terminates.

Diffraction can affect the array far field pattern in two ways: 1) the magnitude of the pressure produced by an individual element can be changed; 2) the phase of the pressure from each element in the array can be changed, and therefore the total array output might not add up to the intended sum for a phased array.

If the baffle is at least a few wavelengths wide and the source of sound is at least a wavelength away from the edge of the baffle, a typical far field pattern of a single source shows small oscillations (about $\pm$ 1 db around the value which one would have if the baffle were infinite) from broadside to about 20° or 30° away from a grazing angle; at a grazing angle the far field is reduced by 6 db, and at some angle (typically near 10°) from grazing the pattern shows an increase of about 2 db (see, for example, Fig. 2 in Ref. 18). Thus diffraction effects due to the finite extent of the baffle can be

[18] V. Mangulis, The Far Field Pattern of a Line Source on a Strip or Half-Plane, TRG, Inc., report No. 023-TM-65-24 (1965).

expected to be important only near grazing angles, unless the phase errors introduced by diffraction are significant also for other angles when one sums the pressures due to the individual sources to obtain the total array pattern.

Let us now consider a typical baffle which terminates, see Fig. 11. For simplicity we assume that the baffle is infinite in the +x-direction, and also in the $\pm z$-directions. We will now replace the actual baffle by an idealized rigid infinite corner baffle of corner angle $\psi > 180^\circ$ as shown in Fig. 11. Of course, one can immediately object to this idealization on the grounds that the choice of ψ is not unique, and therefore in principle the far field calculated on the basis of this idealization is not unique. However, we have shown that the far field for all practical purposes is unaffected by the choice of ψ as long as $\psi > 180^\circ$ if the array is at least a wavelength away from the edge of the corner.[19] One could also object to the idealization because we have replaced a baffle which is finite in the part of the space for which x is negative by another baffle which for $180^\circ < \psi < 270^\circ$ is infinite in that part of the space; however, we have shown that the far field is practically the same for, say, $\psi = 210^\circ$ and for $\psi = 360^\circ$; in the latter case the idealized baffle becomes just a half plane, and the idealized baffle is also finite in the space in which x is negative.[19]

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[19] V. Mangulis, <u>Array on a Corner</u>, TRG, Inc., report No. 023-TN-66-1 (1966).

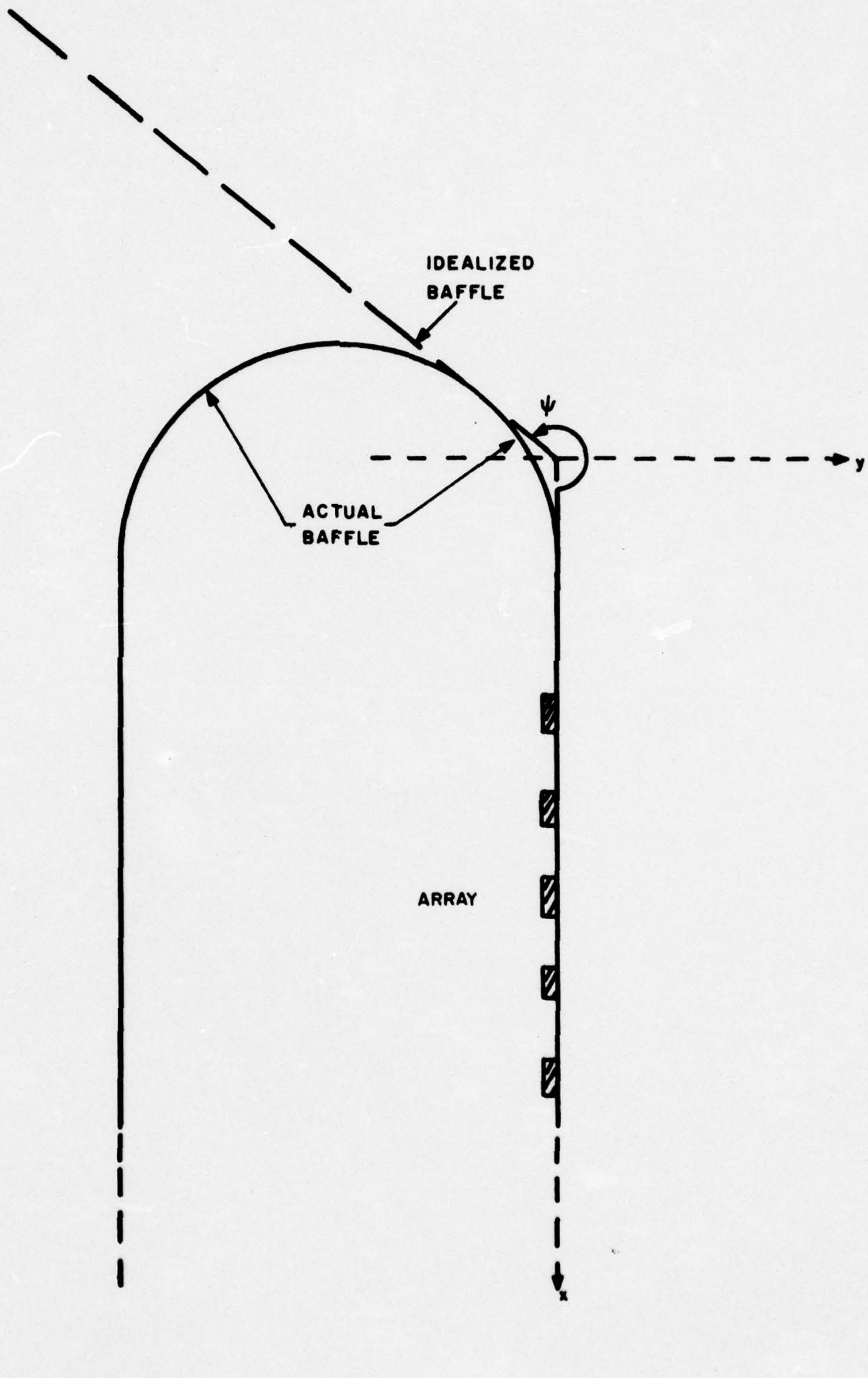

FIGURE II.A.-11 CROSS-SECTIONAL VIEW OF AN ARRAY ON A BAFFLE

The array on the idealized corner baffle is shown in Fig. 12. The far field patterns of an array consisting of 10 rows and 200 columns are shown in Fig. 13 for the main beam steered to 5° from end fire and in Fig. 14 for the main beam steered directly to end fire. The patterns show the expected 6 db loss at a grazing angle ($\phi = 180^{\circ}$) as compared to the patterns for arrays on an infinite plane ($\psi = 180^{\circ}$).

Figs. 13 and 14 and additional calculations presented in Reference 19 show that:

1) diffraction effects become important only when the array is steered to a direction which is at a small angle with respect to the direction in which the baffle is finite;

2) phase differences between array elements introduced by diffraction are negligible if the array is at least a few wavelengths away from the edge of the baffle;

3) the far field pattern for $0^{\circ} \leq \phi \leq 180^{\circ}$ is practically unaffected by the shape of the baffle termination, i.e., by the corner angle ψ;

4) for array steering angles near grazing (i.e., near the direction in which the baffle is finite) the diffraction effects increase the sidelobe levels and may displace the maximum of the pattern.

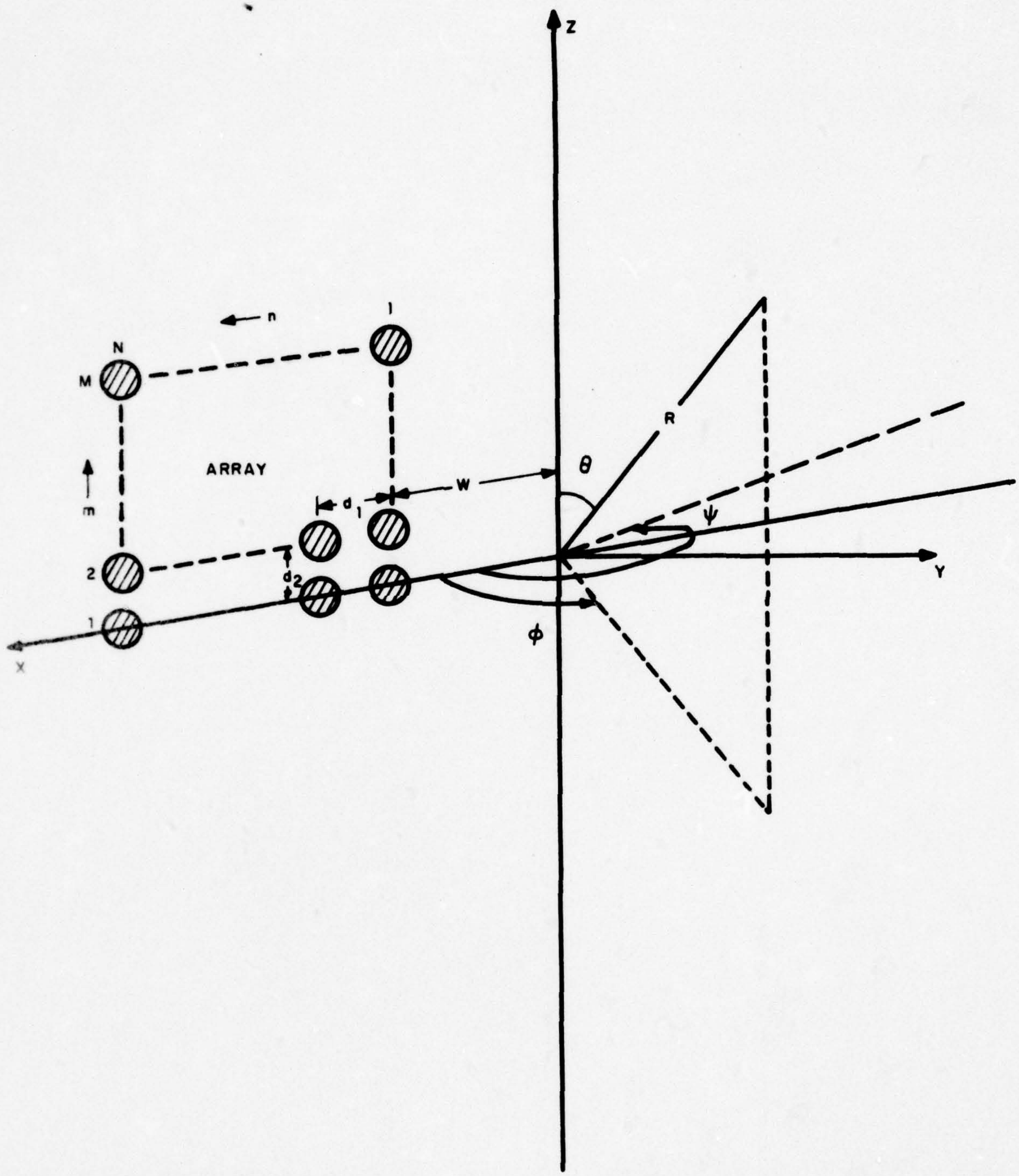

FIG. II.A.-12. ARRAY ON A CORNER

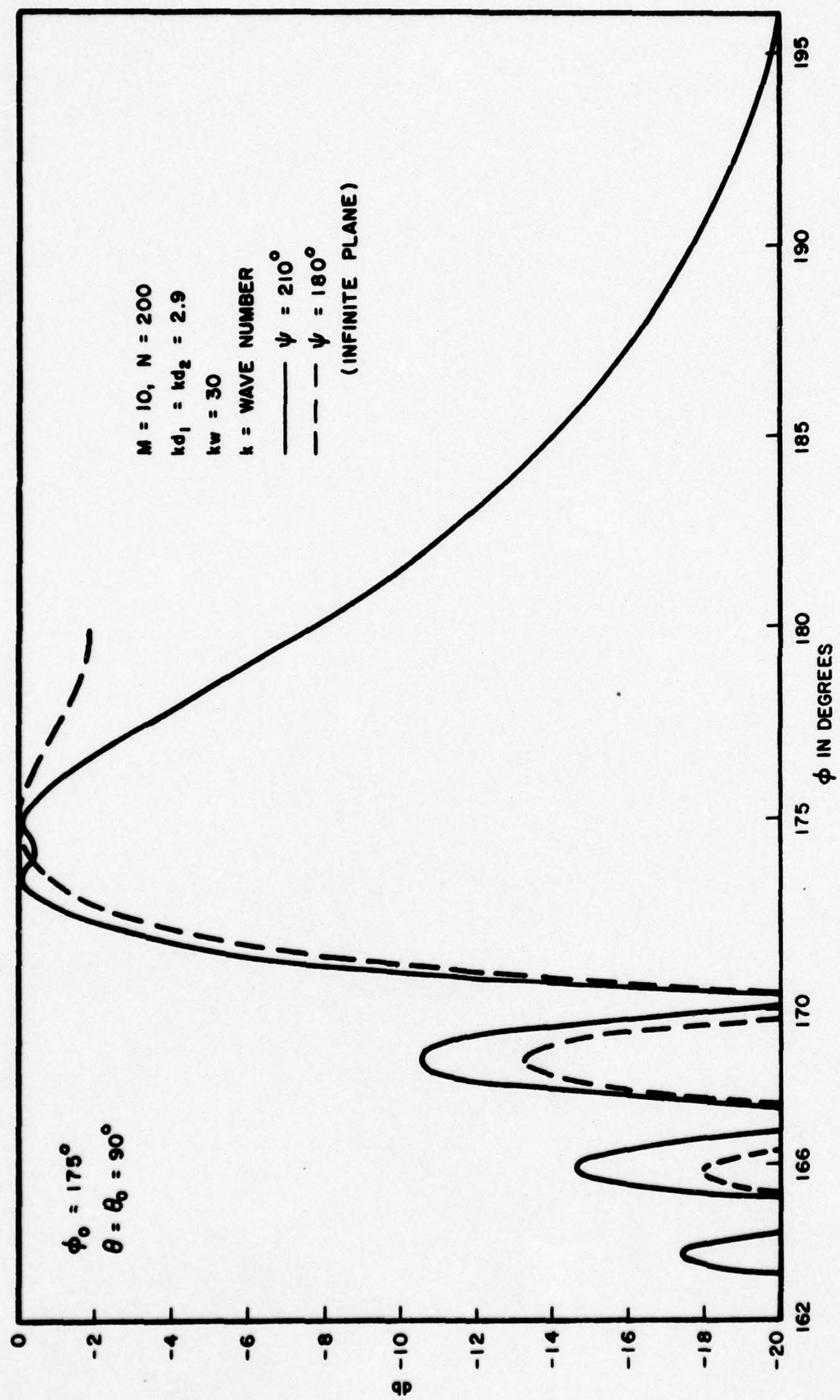

FIGURE II.A.-13. FAR FIELD PATTERN VS ϕ FOR A LONG ARRAY STEERED TO $\phi_0 = 175°$

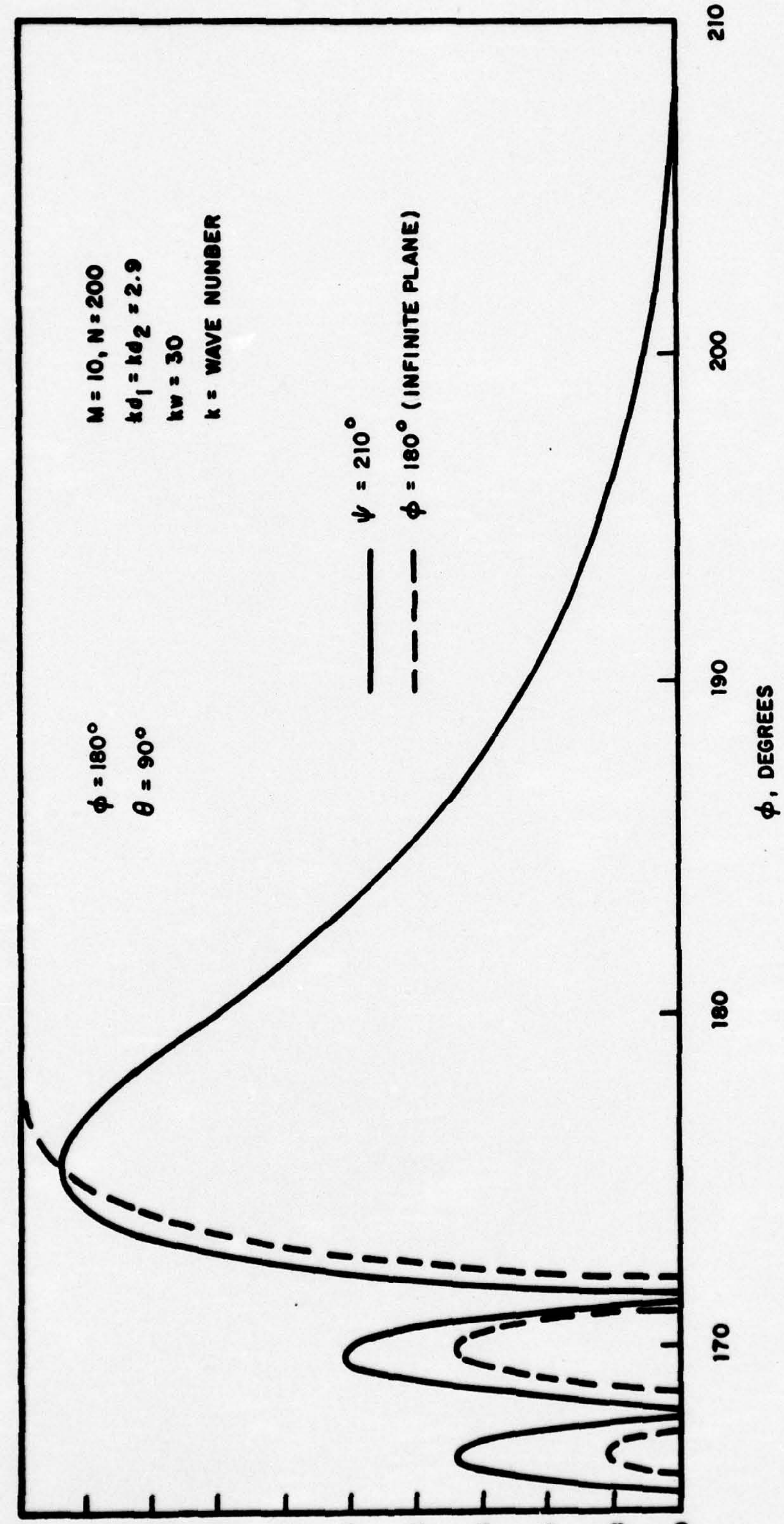

FIGURE II. A.-14 FAR FIELD PATTERN VS. φ FOR A LONG ARRAY STEERED TO $\phi_0 = 180°$

Let us now consider the near field and radiation impedances. As in the previous subsection on the effects of curvature, we can infer that there should not be a significant change in radiation impedances because of diffraction due to the finite extent of baffles because we have shown that if the baffle does not terminate within a few wavelengths, then the piston radiates a field which is almost the same as with an infinite planar baffle, except for some diffusion of the field into the shadow regions. Since the energy which diffuses into a shadow region comes mainly from the adjacent insonified region, and since the shadow regions are at least a few wavelengths away from the piston, the field at the piston remains relatively undisturbed. This argument is confirmed by our calculations for pistons on corners,[7] and by Crane's[20] computations for pistons at the center of a finite disk.

4. Summary

We have investigated the various effects of diffraction on the near field pressure, radiation impedances, and the far field pressure, and we have found that the effects are small, except for a 6 db loss in the far field pressure in the horizontal plane in a direction parallel to the baffle. This 6 db loss becomes important only if the main beam is steered to the same direction (end fire at zero depression

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[20] P.H.G. Crane, The Acoustic Impedance of a Rigid Circular Piston without a Baffle or with a Finite Concentric Baffle, Admiralty Res. Lab. (Britain), Report No. A.R.L./L/N94 (1963), AD 419542.

angle). Thus one can use the much simpler expressions for the array on an infinite rigid plane baffle to calculate the near and far field pressures and the radiation impedances for the actual array on the sonar keel; and the only time a more complicated expression is needed is when one calculates the far field pattern for the main beam phased to end fire in the horizontal plane; this more complicated expression is available and has been programmed for a computer;[19] and, of course, also the simpler expressions for the array on an infinite plane are available and have been programmed for a computer. Consequently, we can evaluate the acoustic radiation from the conformal/planar sonar array for any desired configuration.

B. Transient Effects

The transducer elements in the CONTACT sonar system will be driven in such a way as to sweep the main beam from end fire in one direction to end fire in the opposite direction in a given time interval. One way to achieve this automatic sweep is to drive the array elements with relative phases which change with time. For a linear array the time dependence of such phases turns out to be very simple: the phases have to be equal to a constant times t, where t is the time; when this constant times t is added to ωt, where ω is the common radian frequency at which the array is driven, the resulting time dependence $(\omega + \text{constant})t$ is the same as if the element were driven at a new radian frequency ω + constant. Since the constant is different for different elements, in effect each element

has to be driven at a slightly different frequency to achieve the sweep from end fire to end fire, hence the name "multi-frequency array" was adopted for such an array.

When the array is rectangular, and when the main beam is made to sweep from end fire to end fire in a plane which is not normal to the array, the time dependence of the phases become more complicated, and the array elements have to be driven in a more complicated fashion than just with slightly different, but constant, frequencies; however, we have retained the name "multi-frequency array" to describe the system.

Let us consider the far field radiation patterns of a rectangular array of M rows and N columns. The array is assumed to be in a plane tilted with respect to the horizontal and vertical planes, but the main beam sweeps from end fire to end fire in a plane at a constant depression angle $\Theta = \Theta_o$ from the horizontal plane.

The far field patterns have been calculated[21] as a function of time, and the "multi-frequency array" patterns at some fixed time have been compared to array patterns phased to the same far field direction in the fixed-frequency, steady state case (i.e., when the relative phases between elements do not change with time, and all elements are driven at the same frequency), since these are the patterns for which

[21] V. Mangulis, E. Paige, Far Field Patterns for a Multi-Frequency Array(U), TRG, Inc., report No. 023-TN-65-13 (1965); (Confidential).

simple expressions are available and which have been extensively studied. Fig. 1 shows a typical far field pattern for the arrays steered to end fire at a depression angle of 15°, $M = 6$, and $N = 12$. There are only some insignificant differences in the location and magnitude of some of the sidelobes; the same applies to all other patterns calculated in Reference 21. Since the familiar steady state patterns are much easier to calculate, the computations show that one can safely use the steady state pattern calculations to infer the multi-frequency array patterns.

Consider now the mutual coupling effects in a multi-frequency array.[22] The magnitude of the force exerted by piston A on piston B is not equal to the magnitude of the force exerted by piston B on piston A (the piston velocity magnitudes being equal) if A and B are at different frequencies; the force magnitudes would be equal if the frequencies of A and B were equal. The radiation impedance of a piston in a multi-frequency array changes with time because the phase differences between the velocities of pistons change with time. We are also going to include the effects of switching an array on and switching it off, which introduces other transient effects, because it takes a time d/c for the pressure wave to propagate from one piston to another, if d is the distance between the pistons, and c is the velocity of sound. Similar considerations apply to the near field pressure.

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[22] V. Mangulis, Transient Mutual Coupling Effects in a Multi-Frequency Array(U), TRG, Inc., report No.023-TN-65-1(1965); (Confidential).

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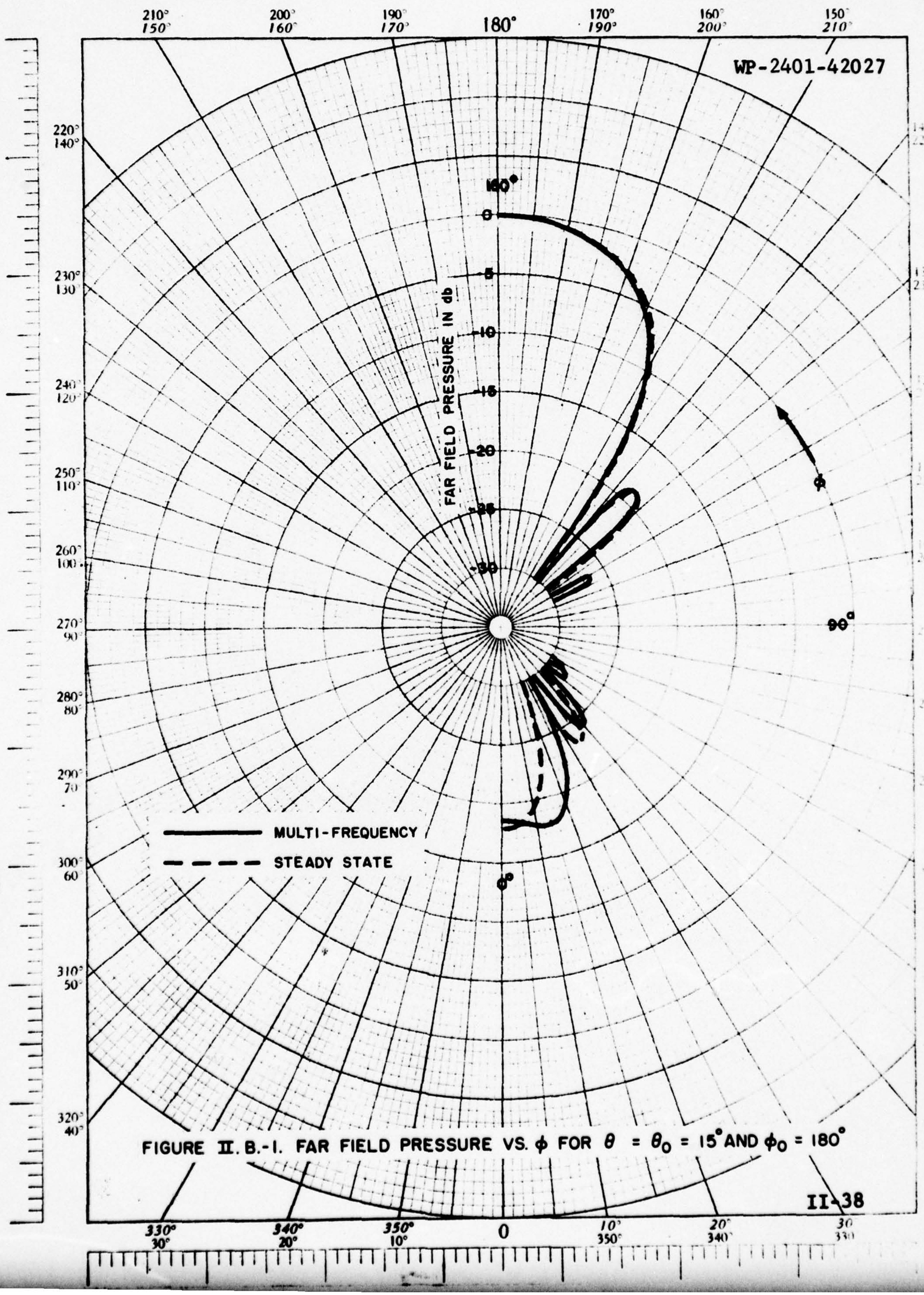

FIGURE II. B.-1. FAR FIELD PRESSURE VS. ϕ FOR $\theta = \theta_0 = 15°$ AND $\phi_0 = 180°$

The time-dependent radiation impedances have been computed by the use of an exact Fourier series expansion and also by the use of an approximation based on a simplified physical model; the agreement between both computations has been good, therefore the simpler calculations based on the simplified physical model (described below) can be used instead of the more complicated Fourier series expansion.[22]

It can be shown[23] that the sound field of a circular piston in an infinite rigid plane baffle is of the form shown in Fig. 2, i.e., the sound field occupies a region in space which is obtained by drawing spheres of radius $R = c(t - T + \tau_n)$ from each point on the n^{th} piston, if the piston is switched on at $t = T - \tau_n$. Since the signal from any part of the piston travels with a velocity c, the field contains a transient envelope (shown shaded in Fig. 2) in which the acoustic signals from some parts of the piston have not yet arrived; outside this transient region there is no field due to the n^{th} piston, and inside this transient region a steady state has been reached.

Thus the time-dependent self-impedance of a circular piston is transient for a time interval $2b/c$, and thereafter is the same as in the steady state; the self-impedance has been evaluated numerically during the time interval $2b/c$.[23]

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[23] V. Mangulis, The Time-Dependent Force on a Sound Radiator Immediately Following Switch-On, TRG, Inc., report No. 023-TN-64-2 (1964); also published in Acustica, vol. 17, p. 223 (1966).

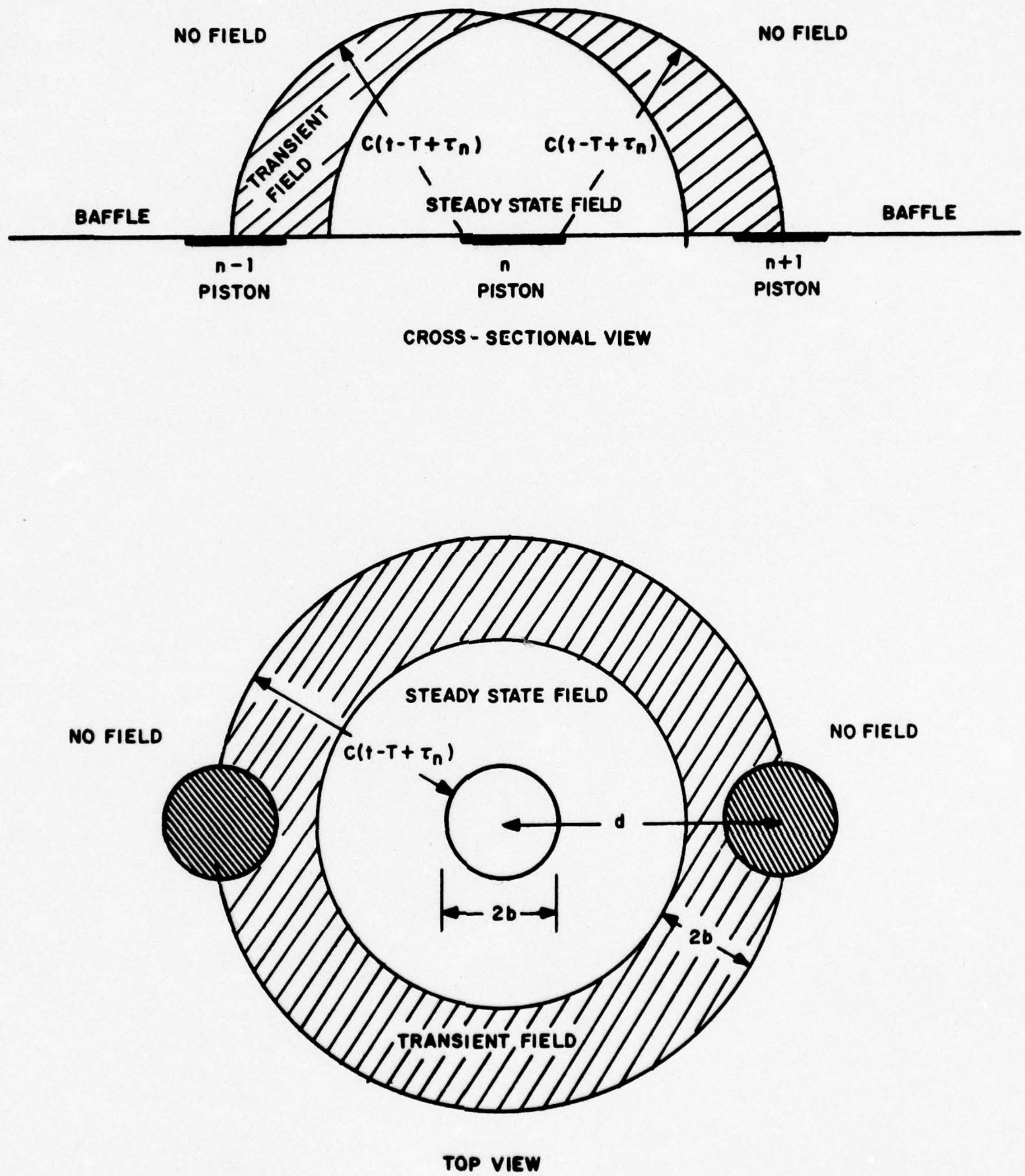

FIGURE II B-2 THE SOUND FIELD OF THE n^{th} PISTON AT TIME t, SWITCHED ON AT $T-\tau_n$

Consider a linear array of pistons, spaced a distance d apart. The force exerted by one piston on another (or the corresponding mutual coupling coefficient) is also time dependent; the transient field arrives at the edge of the $m^{\underline{th}}$ piston a time $(|m-n|d-2b)/c$ after the $n^{\underline{th}}$ piston has been turned on, the force is transient for a time interval $4b/c$, and it becomes the same as in the steady state at the time $(|m-n|d + 2b)/c$. It is obvious that the transient mutual coupling force starts from zero at $(|m-n|d-2b)/c$; for pistons with dimensions less than a wavelength the transient force will be of the same order as in the steady state or less because the physical meaning of the transient state is simply that only a part of one piston exerts a force on a part of another piston since the field from the whole piston has not had time to arrive yet. Thus the mutual coupling coefficient during the transient state is approximately the same as for pistons with smaller areas, i.e., those areas which exert forces on each other.

The mutual coupling force is transient for a time interval $4b/c$ which in our application might be of the order of 2×10^{-4} seconds, a rather small quantity. To simplify the calculations we can replace the actual situation described above by the following simplified physical model: the force on the $m^{\underline{th}}$ piston due to the $n^{\underline{th}}$ piston is zero until the time $|m-n|d/c$ after the n^{th} piston has been turned on, and it is the same as in the steady state thereafter. The error which we introduce in the total radiation impedance (taking all

pistons into account) will be of the order of one mutual coupling coefficient or less, because for the linear array at the most roughly one mutual coupling force is transient at any given time, and the magnitude of the transient force is of the same order as the force in the steady state or less. The radiation impedances will be exact after a time $t = T = \tau_N + (N-1)d/c$ because then all pistons will have been turned on, and the wave from the last piston (the $N^{\underline{th}}$) to be switched on will have reached all the other pistons.

As mentioned above, this approximation agrees well with the Fourier series expansion.

Let us now compare the radiation impedance of the $m^{\underline{th}}$ element for the multi-frequency array at some fixed time $\left[\text{denoted by } \rho c\pi b^2(R_m{}^s + iX_m{}^s)\right]$ with that for the steady state array steered to the same angle $\left[\text{denoted by } \rho c\pi b^2(P_m + iQ_m)\right]$; ρ is the water density, c the sound velocity, b the radius of the piston.

Tables I-IV show the radiation impedances for some selected elements m and a few steering angles ϕ_o; $d = 0.171$ meters, $b = 0.057$ meters, frequency of the first element 4000 cps, total number of elements 199.

While the transient radiation impedances of elements in a multi-frequency array (shown here and in Reference 22) are different from those in a steady state single-frequency array, they are not radically different, and therefore the designer of transducer elements is not faced with any new problems when designing for the multi-frequency array. For example, if element velocity control in the array is

Table II. B.-I

f_1 = 4000 cps, $\phi_o = 90^\circ$ (broadside)

m	Radiation resistances		Radiation reactances	
	R_m^S	P_m	X_m^S	Q_m
1	0.407 *	0.408	0.541	0.542
10	0.437	0.438	0.464	0.464
50	0.437	0.438	0.457	0.458
100	0.439	0.440	0.459	0.459
150	0.439	0.438	0.459	0.458
190	0.439	0.438	0.465	0.464
199	0.409	0.408	0.542	0.542

*Largest percent difference $= \left[\left| P_1 - R_1^S \right| / \frac{1}{2} (P_1 + R_1^S) \right] \times 100 \approx 0.4\%$

Table II. B.-II

f_1 = 4000 cps, $\phi_o = 30^\circ$

m	Radiation resistances		Radiation reactances	
	R_m^S	P_m	X_m^S	Q_m
1	0.534	0.536	0.756	0.757
10	0.392	0.392	0.737	0.740
50	0.398	0.398	0.745	0.747
100	0.401	0.399	0.745	0.743
150	0.397	0.396	0.747	0.744
190	0.436	0.435	0.751 *	0.740
199	0.240	0.240	0.617	0.613

*Largest percent difference = 1.6%

Table II. B.-III

$f_1 = 4000$ cps, $\phi_o = 15^o$

m	Radiation resistances		Radiation reactances	
	R_m^S	P_m	X_m^S	Q_m
1	0.543	0.545	0.923	0.923
10	0.397	0.399	0.955	0.957
50	0.391	0.397	0.963	0.959
100	0.403	0.407	0.969	0.949
150	0.414 *	0.399	0.945	0.922
190	0.292	0.306	0.994	0.982
199	0.219	0.220	0.655	0.649

*Largest percent difference = 3.7 %

Table II. B.-IV

$f_1 = 4000$ cps, $\phi_o = 5^o$

m	Radiation resistances		Radiation reactances	
	R_m^S	P_m	X_m^S	Q_m
1	0.565	0.567	1.242	1.244
10	0.408	0.417	1.300	1.290
50	0.315	0.367	1.329	1.293
100	0.235 *	0.324	1.313	1.280
150	0.192	0.261	1.229	1.217
190	0.194	0.211	1.027	1.020
199	0.213	0.214	0.671	0.664

*Largest percent difference = 31.6 %

achieved by making the internal transducer impedance magnitude large as compared to the radiation impedance magnitude,[1] a transducer designed for a steady state single-frequency array will also work in the multi-frequency array because the radiation impedance magnitudes in the two arrays are comparable.

Recently Sherman and Moran[24] have also considered the transient near field of an array; they conclude that cavitation is not likely to be initiated by transient effects.

In summary: while transient effects can be noticed in the far field patterns and in radiation impedances, they are of minor importance. The effects can be easily evaluated by the use of the existing theory[21,22,23] for the conformal/planar sonar array for any desired configuration, if considered to be necessary, but the calculations already performed indicate that transient effects have little practical significance.

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[24] C.H. Sherman and D.A. Moran, Transient Sound Field of Simple Arrays of Circular Pistons, Parke Mathematical Laboratories, Inc., report No. 0172-SR-1 (1966); AD 644539.

III. BAFFLE DESIGN

In 1961 TRG showed[1] that the observed inability to steer a planar sonar array to end fire is caused by the loss in pressure at the hull of the ship at grazing angles due to the finite rigidity of steel plates of practical thickness. This theoretical deduction was later supported by experimental measurements.[2] TRG suggested[1] that the use of mechanical resonators (also called stiffeners) fastened to the inner surface of the hull could increase the impedance of the hull to a satisfactory level over a bandwidth sufficient for practical purposes. The effectiveness of resonators was later verified experimentally.[2,3,4,5]

Obviously the design of the baffle is just as important as the design of the transducer elements if one expects to phase the array

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[1] J. Lurye, V. Mangulis, W. Graham, A Study of Conformal Arrays for Long Range Sonars, Phase One, TRG-142-TR (1961); AD 323262(Confidential).

[2] V. Mangulis, S. Gardner, A. Novick, W. Graham, A Study of Conformal Arrays for Long Range Sonars, Phase Two, TRG-142-TR-2 (1963); (Confidential); Vol. III, AD 336258, Appendix XI.

[3] D. Yarmush, S. Aronson, Experimental Study of Mechanical Resonators Attached to Bars, TRG-142-TN-64-11 (1964).

[4] S. Gardner, D. Chase, Noise and Vibration Measurements on USS Brownson (DD 868), TRG-142-TN-64-4 (1964); (Confidential).

[5] J. Lyons, T. DeFilippis, Results of the Dodge Pond Test Program on a 6 x 12 Element Conformal Array, TRG-142-TN-64-5 (1964); (Confidential); AD 377882.

to end fire. The only method to rigidize a baffle, which has been proven both theoretically and experimentally, is the approach proposed by TRG, i.e., the attachment of mechanical resonators or stiffeners to the inner surface of the sonar keel.

We describe below some of the recent theoretical and experimental work in connection with the design of rigid baffles.

A. Theoretical Results

Theoretical studies of the effectiveness of resonators have considered single resonators,[6] resonators distributed uniformly and continuously[7] behind an infinite elastic plate, and an infinite set of resonators or parallel stiffeners[8] distributed at regular intervals behind an infinite plate in contact with water; the latter study will be described in detail here. The results for this discrete distribution approach the results for the uniform and continuous distribution when the spacing between resonators approaches zero. The purpose of this theoretical study is to find the maximum spacing between resonators at which the plate is still sufficiently rigidized. We will calculate the pressure at the plate due to an incident plane wave; the plate will be considered to be sufficiently rigidized when this pressure at the plate does not differ appreciably from the pressure at an ideally rigid

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[6] D. Yarmush, Theoretical Study Resonant Stiffening Devices, TRG-142-TN-64-10 (1964).

[7] Reference 2, Appendix XIII.

[8] D. Yarmush, V. Mangulis, Acoustic Pressure at an Infinite Plate with Parallel Stiffeners, TRG-023-TM-67-8 (1967).

plate. It has been shown experimentally that a plate can be rigidized with practical spacings between resonators;[2,5] however, we may have used smaller spacings and more resonators than necessary, which motivates this study. One purpose of this work is to provide a more thorough analytic foundation for the design of resonant stiffeners.

Consider an elastic plate of thickness h in contact with water occupying the half-space $x \leq 0$, see Figure 1. Resonators or parallel stiffeners are attached to the plate at $ns-b < y < ns$, $n = 0, \pm 1, \pm 2 \ldots$, where b is the width of the resonator attachment (assumed to be very small as compared to a wavelength in water or flexural wavelength in the plate), and s is the spacing between resonators. The resonators are assumed to be infinite in the z-direction (normal to the plane of the paper in Fig. 1).

The effects of resonators are given by specifying: 1) the force which the resonator exerts on the plate at the attachment point; this force is proportional to the plate displacement; 2) the moment which the resonator exerts on the plate; this moment is proportional to the slope of the plate at the resonator attachment point. Various designs of resonators are possible; for example, a simple and effective device used frequently consists of a short stud, which acts as a spring, with a larger piece, which acts as a mass, at the end. An infinite number of different resonator designs will give the same force and moment at the attachment point; therefore for simplicity in the calculations presented below we assumed that the resonator forces and moments are

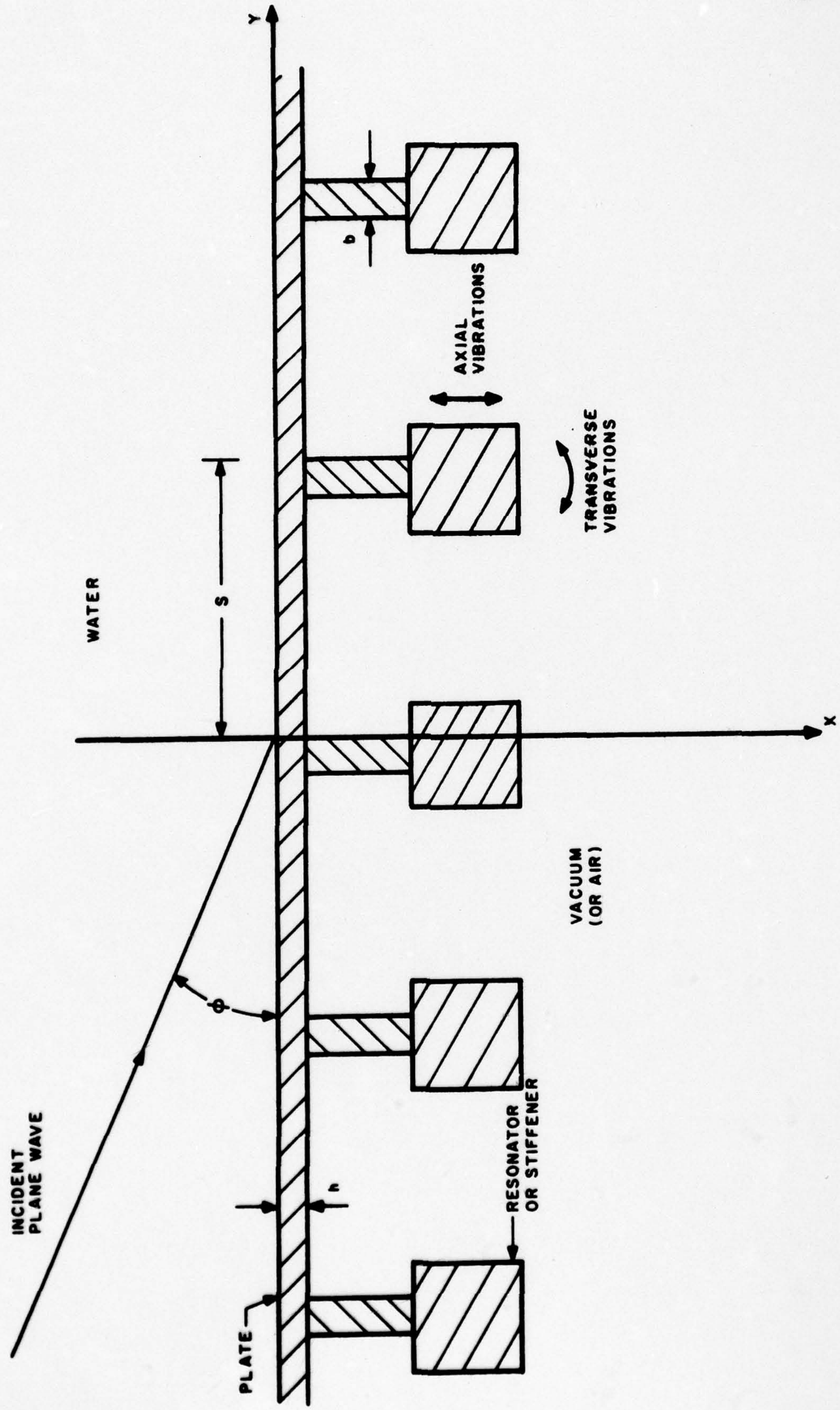

FIG. III.A.-1 PLATE WITH RESONATORS

1/10 of those for a simple steel rod of circular cross section with the first axial and second transverse vibration resonance at 2500 cps. While the dimensions, which are determined by the resonant frequency, of such a simple resonator then must be very large (length 42 cms, radius 6 cms), one should remember that an infinite number of other designs will give the same forces and moments, and a more efficient resonator (such as the stud plus mass mentioned above) will be much smaller in size.[7] Note also that we are actually considering a resonator only 1/10 of the size of the rod, since the forces and moments are assumed to be 1/10 of those for the rod. Most calculations were performed with the 1/10 simple rod resonator only because the mathematical expressions for forces and moments were easy to obtain and easy to compute.

Figure 2 shows the average pressure magnitude on the plate vs. the spacing between resonators. The plate is 1/4 inch thick, and the frequency is 2500 cps. An approximation in which the resonators are assumed to be distributed uniformly and continuously[7] behind the infinite plate gives 0 db for the pressure for all spacings and angles in Fig. 2. The pressure reduction for $s \geq 7$ cms is easily understood: the wavelength for free flexural vibrations is 14 cms (for $f = 2500$ cps, $h = 1/4$ inch), therefore stiffeners placed at half-wavelength intervals ($s = 7$ cms) cannot be very effective. While we have plotted the average pressure in Fig. 2 (and also in the subsequent figures) instead of the actual pressure distribution on the plate, the actual distribution rarely differs by more than 0.01 db from the average pressure, unless the pressure levels are very low.

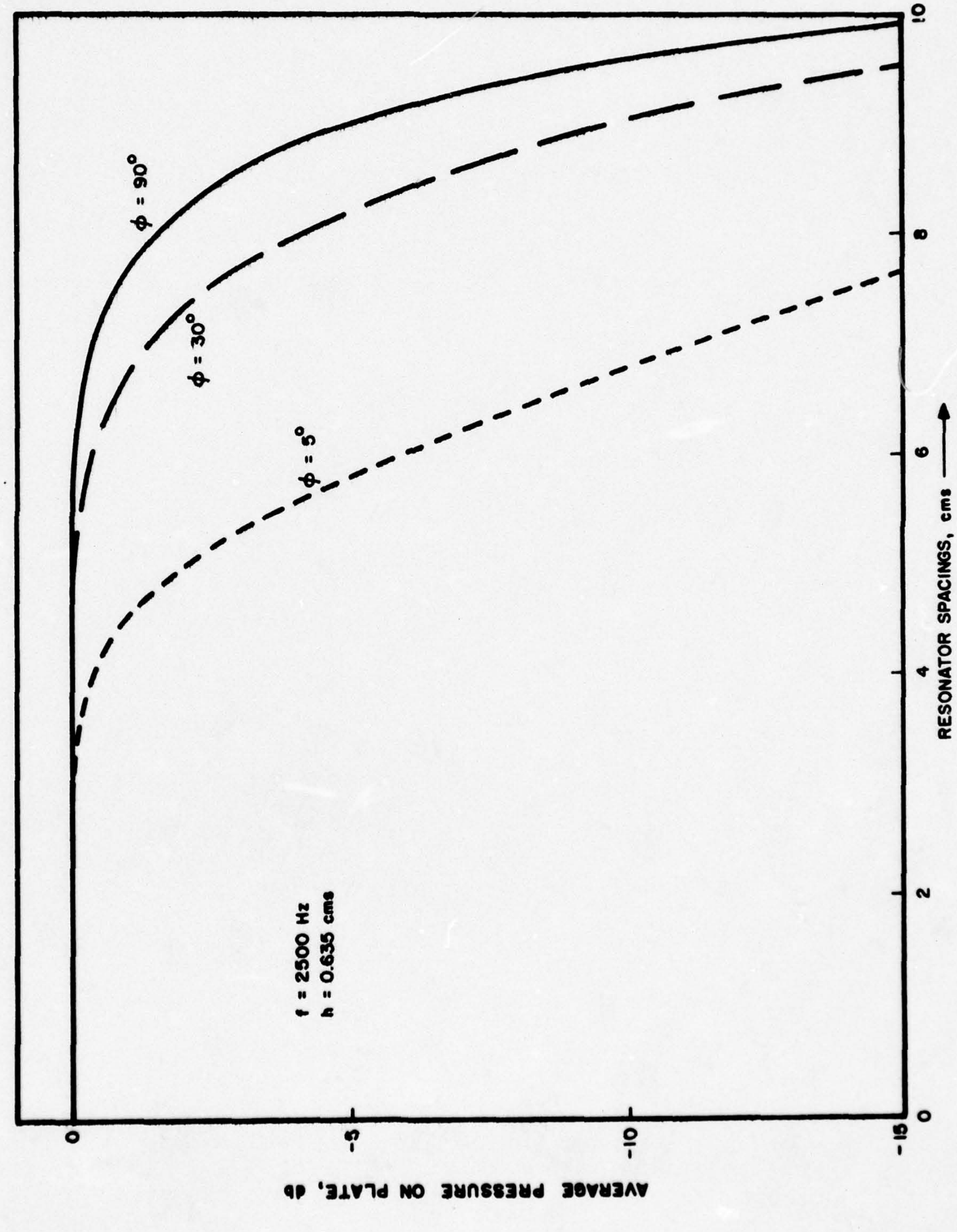

FIGURE III. A.-2. AVERAGE PRESSURE ON THE PLATE VS. THE SPACING BETWEEN RESONATORS; 1/4 INCH PLATE

Figure 3 shows the average pressure vs. the angle of incidence ϕ for resonator spacings $s = 4, 5$, and 6 cms. Figures 2 and 3 indicate that a resonator spacing of 4 cms is acceptable; the pressure is reduced by less than 2 db for $\phi \geq 2^{\circ}$.

Figure 4 shows the average pressure vs. the frequency. The corresponding resonator or stiffener strength vs. the frequency is shown in Table I, where F and G are quantities proportional to the force and moment exerted by the resonator.[8] Although the stiffener itself exerts the maximum force and moment on the plate at 2500 cps, as shown in Table I, at $\phi = 5^{\circ}$ Figure 4 shows that the plate is most rigid between 2200 cps and 2400 cps, and the pressure is lower at 3000 cps than at 2000 cps. Consequently, an improvement in the pressure level in a total bandwidth of 1000 cps could be obtained by either shifting this band to lower frequencies (from 1800 cps to 2800 cps) with a stiffener resonance at 2500 cps as in Figure 4, or by changing the stiffener resonance frequency to 2700 cps and keeping the band from 2000 cps to 3000 cps. In either case one would not lose more than 2 db in pressure at the extreme frequencies of the band with the stiffener spacing $s = 4$ cms and $\phi \geq 5^{\circ}$.

Figure 5 shows the average pressure vs. the thickness of the plate for a resonator spacing of 4 cms.

Figure 6 shows the average pressure vs. the resonator strength, where we have fixed the relative values of F and G in such a way that $G = -2 \times 10^{-3}$ F. For small values of F and G the pressure

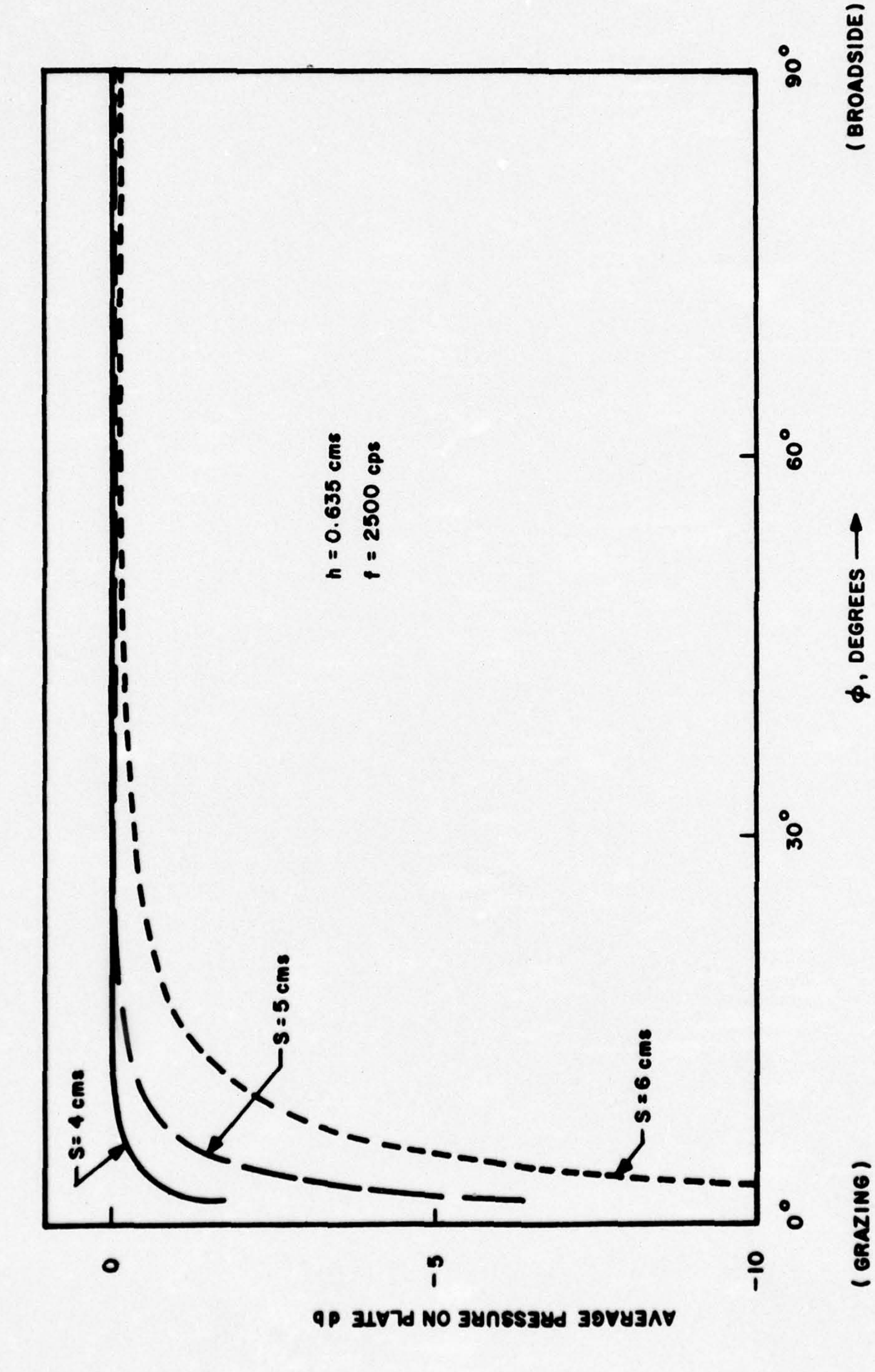

FIGURE III. A-3. AVERAGE PRESSURE ON THE PLATE VS. ANGLE OF INCIDENCE; 1/4 INCH PLATE

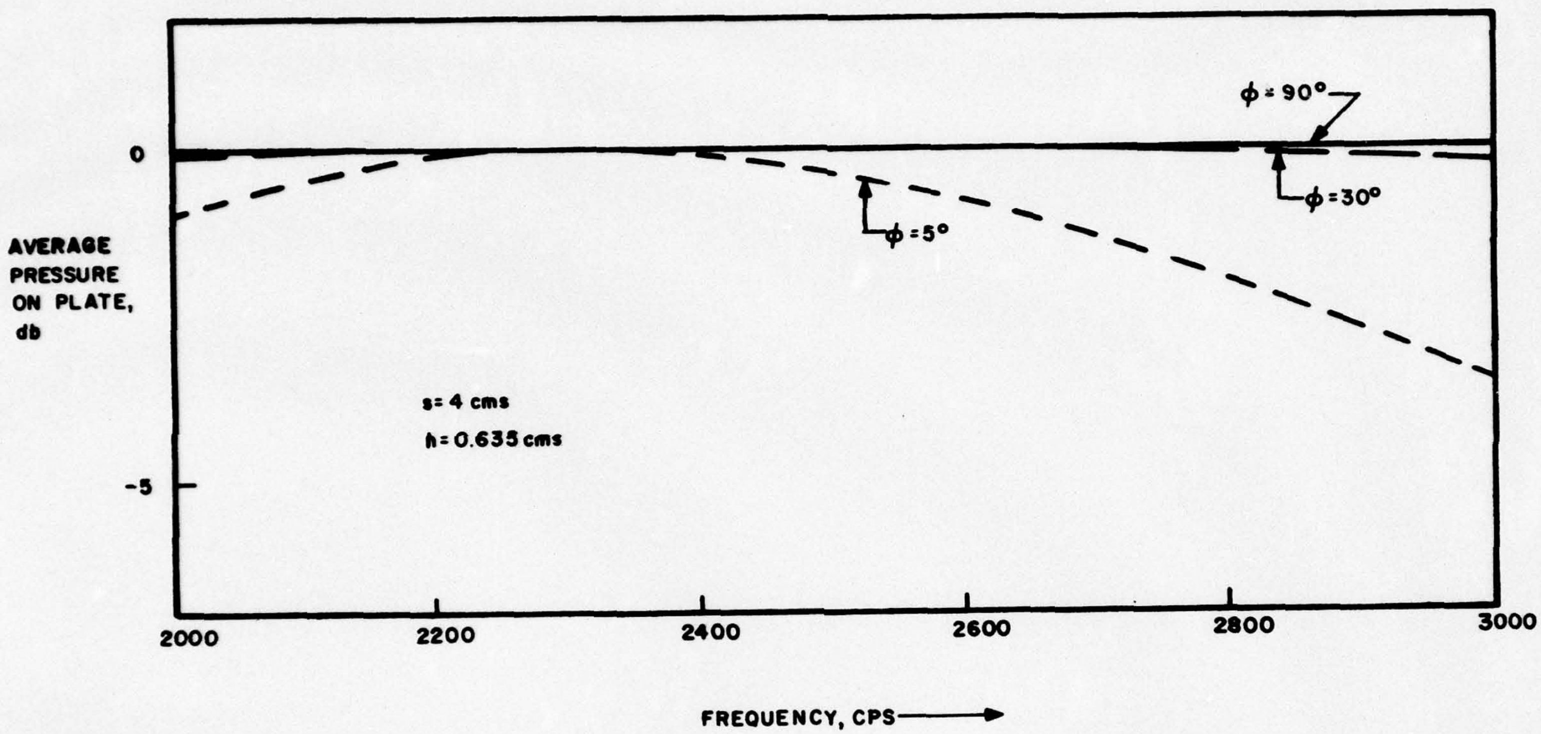

FIGURE III A-4 AVERAGE PRESSURE ON THE PLATE VS. FREQUENCY; 1/4 INCH PLATE

TABLE III. A.-I

$\frac{1}{10}$ of Resonator or Stiffener Strength vs. Frequency

Frequency cps	F, 10^{10} newtons/m^2	G, 10^7 newtons
2000	1.14	-0.85
2100	1.51	-1.49
2200	2.10	-2.52
2300	3.23	-4.48
2400	6.30	-9.65
2500	46.83	-64.71
2600	-9.49	19.01
2700	-4.48	9.34
3800	-2.99	6.58
2900	-2.27	5.26
3000	-1.84	4.48

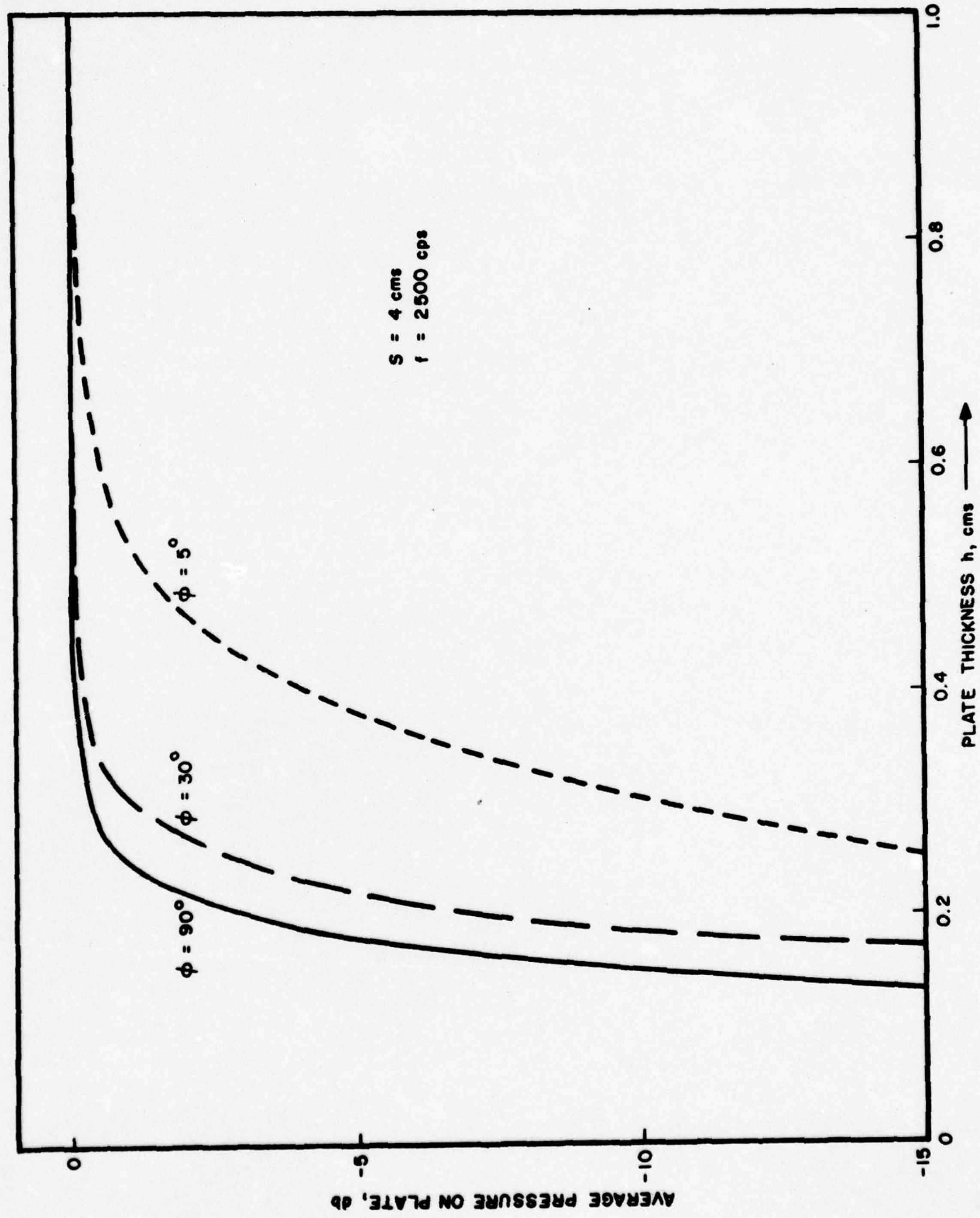

FIGURE III.A.-5. AVERAGE PRESSURE ON THE PLATE VS. THE THICKNESS OF THE PLATE

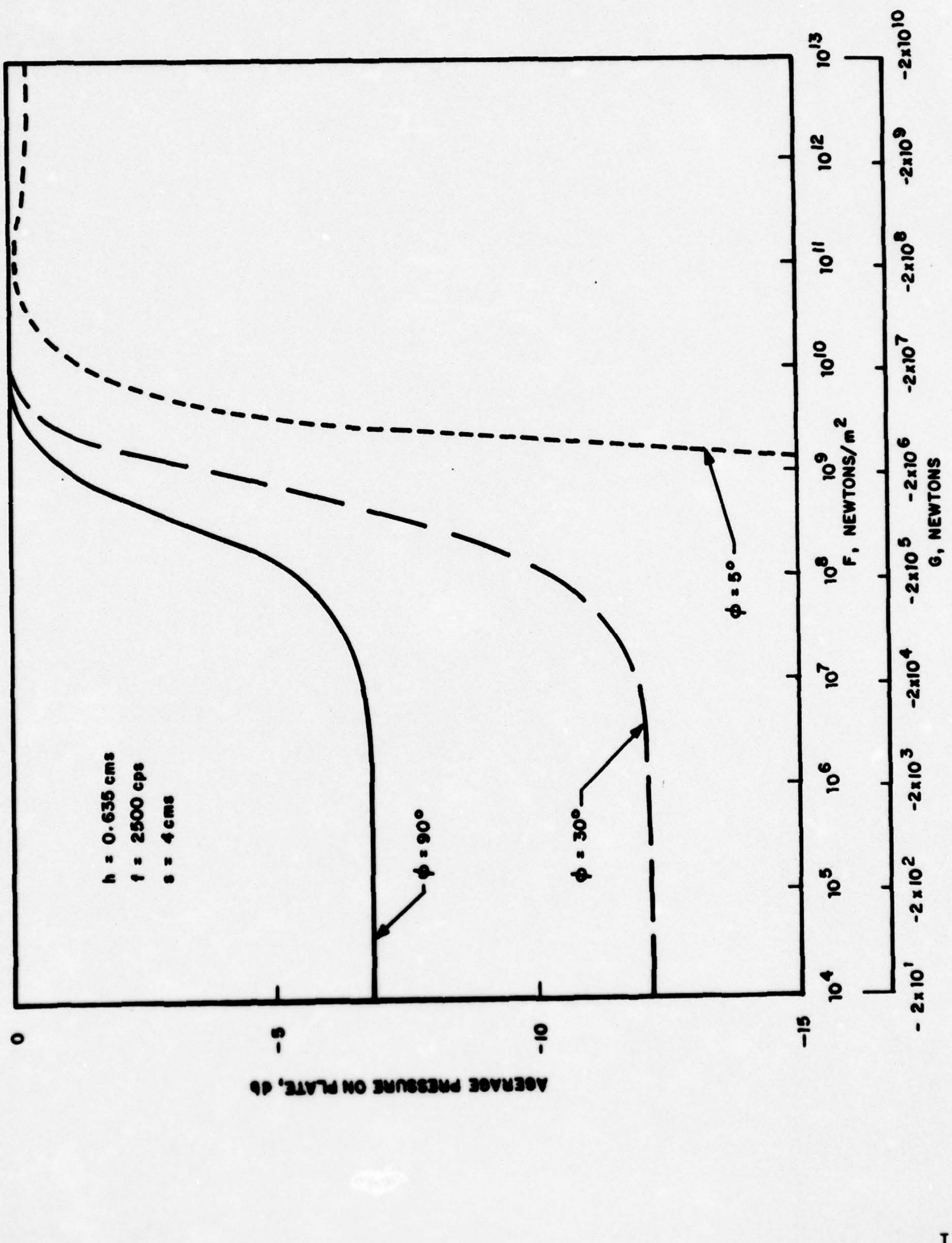

FIGURE III A.-6. AVERAGE PRESSURE ON THE PLATE VS. THE RESONATOR STRENGTH; 1/4 INCH PLATE

magnitudes approach the values for a 1/4 inch thick plate without resonators, which are:

$$-27.0 \text{ db at } \phi = 5^{o},$$

$$-12.1 \text{ db at } \phi = 30^{o},$$

$$-6.8 \text{ db at } \phi = 90^{o},$$

For a 1/2 inch thick plate without resonators the pressure levels are:

$$-21.1 \text{ db at } \phi = 5^{o},$$

$$-6.9 \text{ db at } \phi = 30^{o},$$

$$-2.9 \text{ db at } \phi = 90^{o},$$

Figures 7-10 show the same data for a half inch thick plate as in the previous figures for a quarter inch thick plate.

Since the wavelength for free flexural vibrations for the 1/2 inch thick plate is 20 cms, the pressure vs. resonator spacing in Fig. 7 shows a loss at the half-wavelength spacing, $s = 10$ cms. Very small losses are shown in Fig. 8 at grazing angles, which should be comparedwith Fig. 3 for the 1/4 inch thick plate. Figures 7 and 8 indicate that at $f = 2500$ cps a resonator spacing of 6 cms for the 1/2 inch thick plate is acceptable.

However, if we now compare Figures 9 and 4, which show pressure vs. frequency at a resonator spacing of 4 cms, we do not see any improvement with the thicker plate. If we compare Figures 10 and 6, which show pressure vs. the resonator strength, we see that the thicker plate has a higher pressure level at low resonator strengths and at

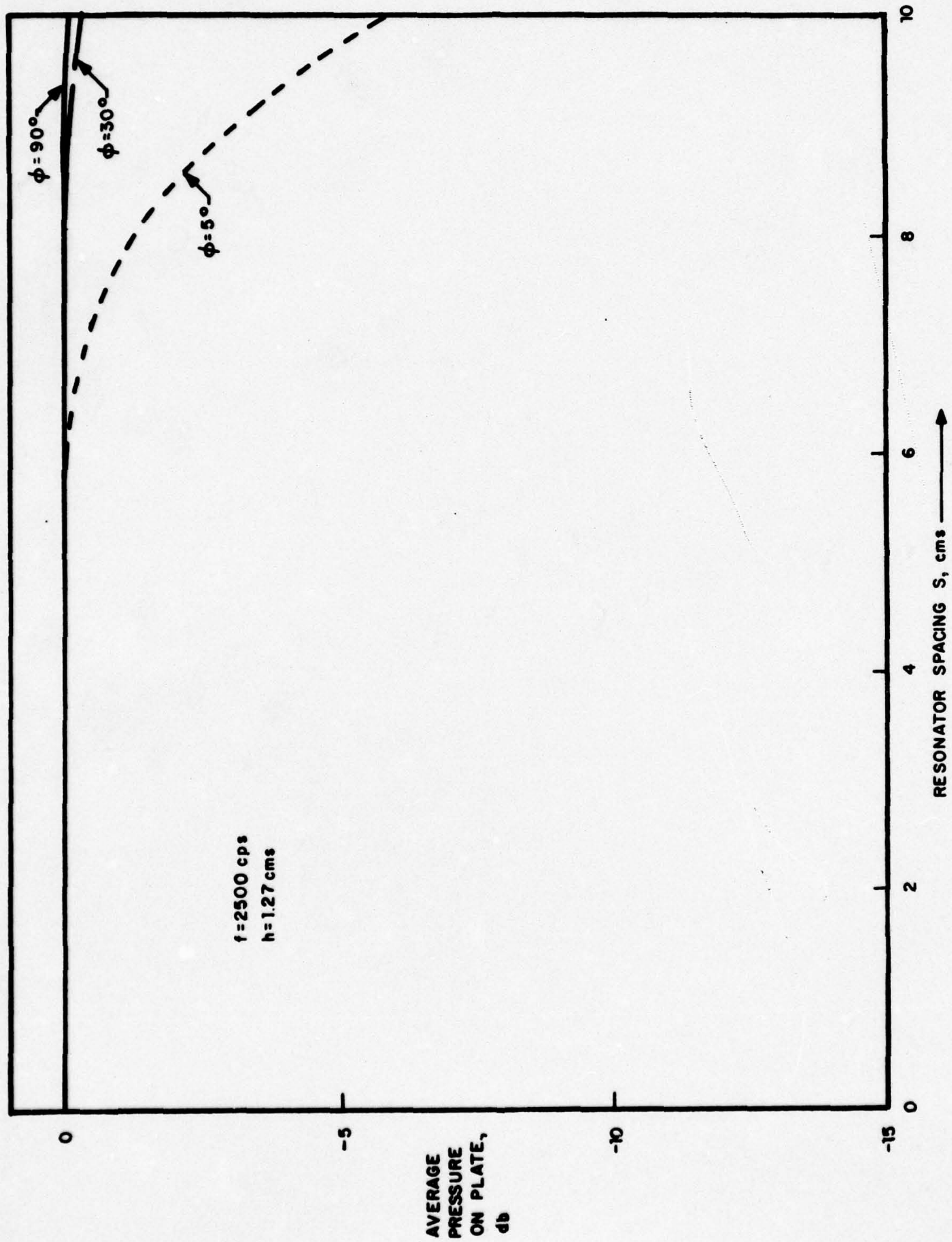

FIGURE III.A-7. AVERAGE PRESSURE ON THE PLATE VS. THE SPACING BETWEEN RESONATORS; 1/2 INCH PLATE

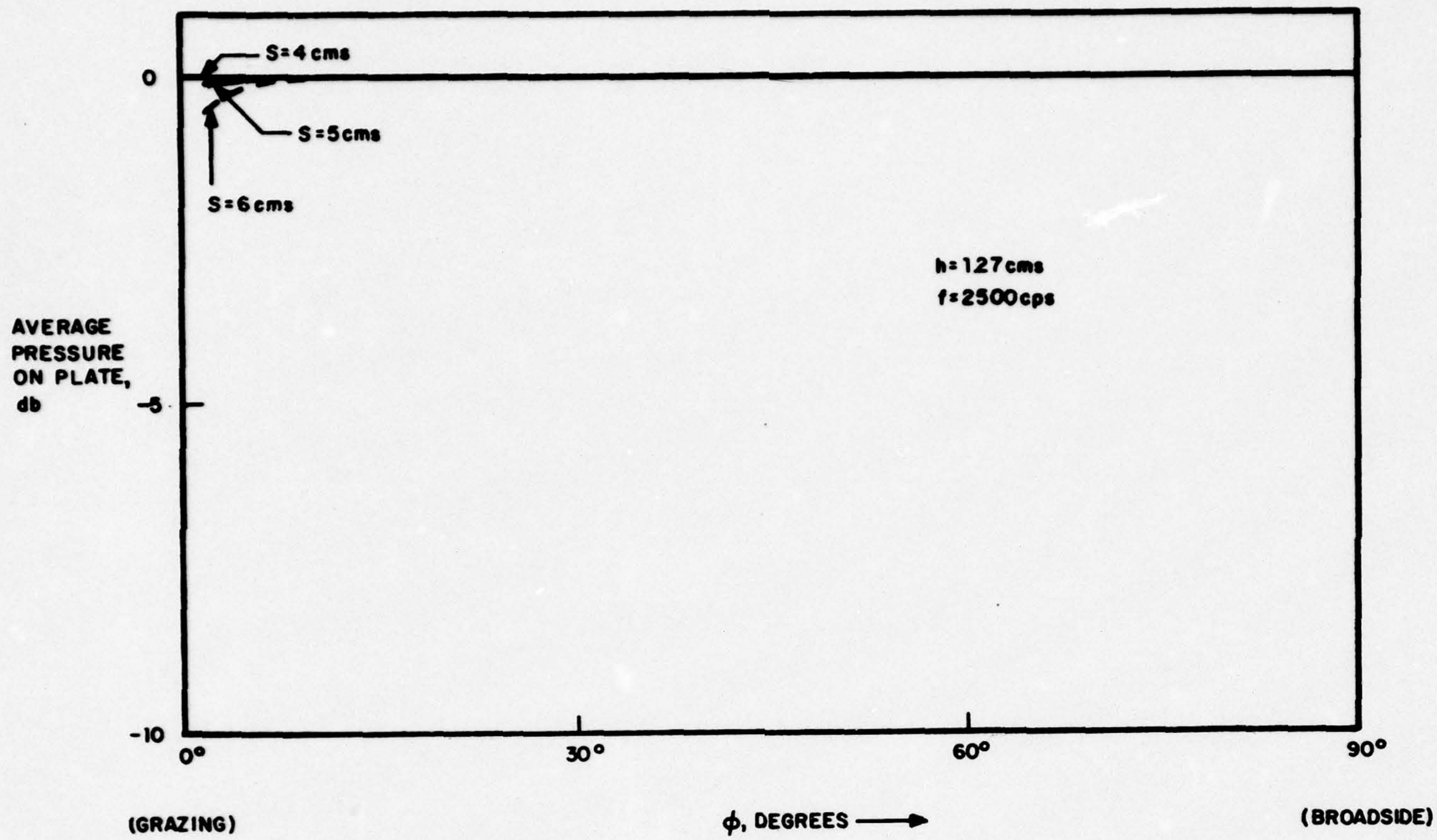

FIGURE III.A.-8. AVERAGE PRESSURE ON THE PLATE VS. THE ANGLE OF INCIDENCE; 1/2 INCH PLATE.

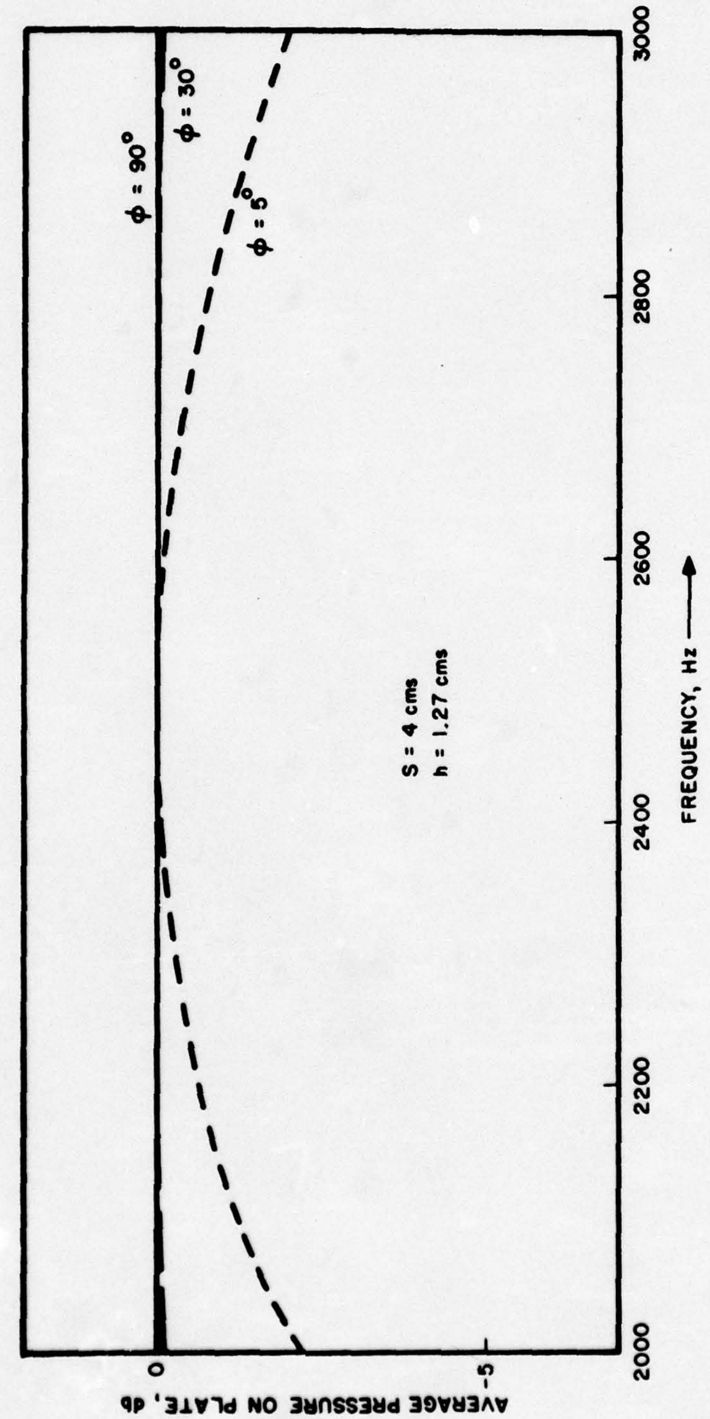

FIGURE III. A.-9 AVERAGE PRESSURE ON THE PLATE VS. FREQUENCY; 1/2 INCH PLATE

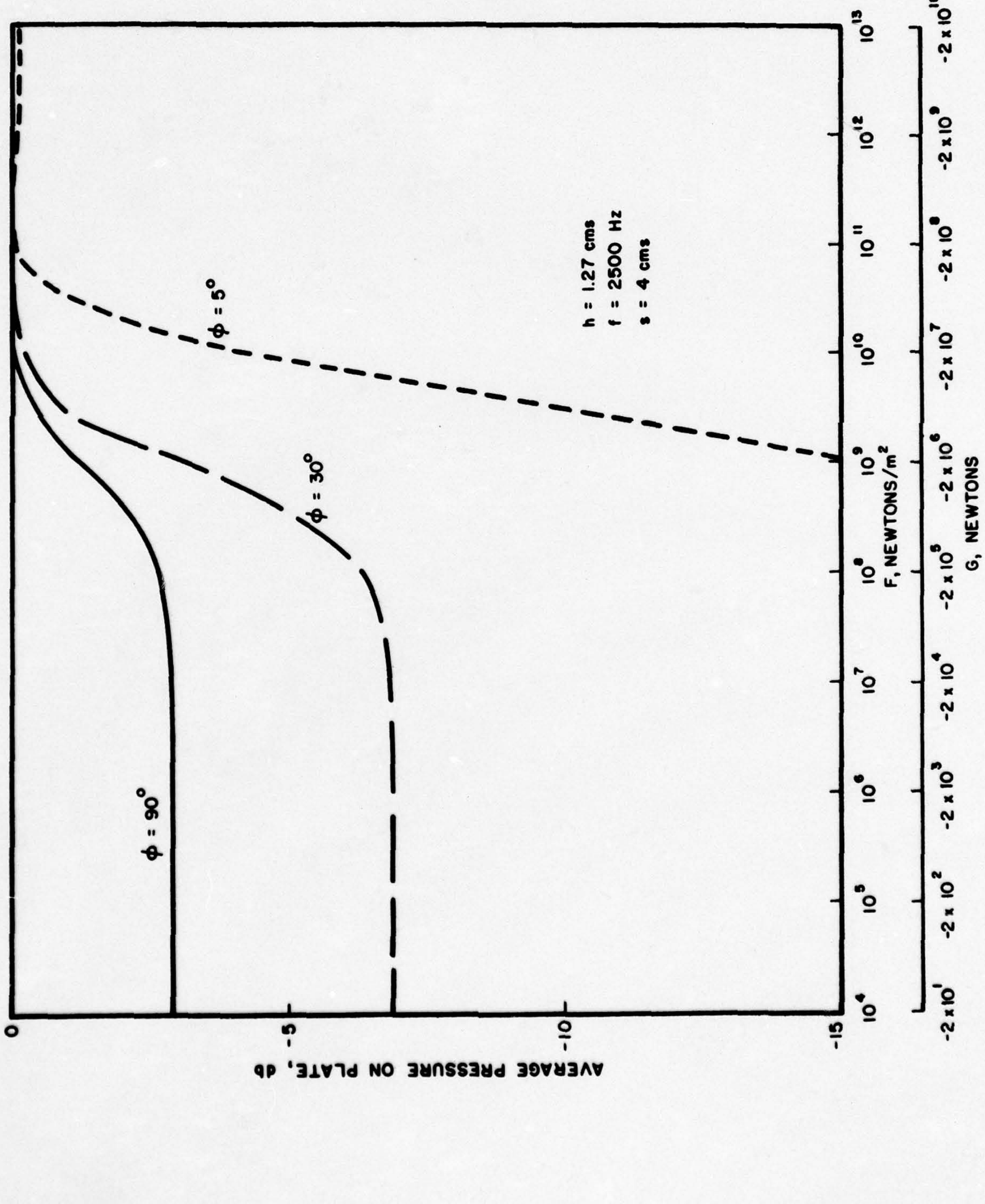

FIGURE III.A.-10. AVERAGE PRESSURE ON THE PLATE VS. THE RESONATOR STRENGTH; 1/2 INCH PLATE

high resonator strengths, but that at intermediate resonator strengths, where the transition from a flexible to a rigid plate takes place, the slope of the pressure curve is steeper for the thinner plate, and as one increases the resonator strength, the thinner plate becomes rigidized sooner than the thicker plate. Apparently the rigidization of the plate depends on the ratio of the plate mass to the resonator strength. This explains the lack of improvement in Figure 9 as compared to Figure 4, because at the extreme frequencies of the band the resonator strengths are lower than in the middle of the band, see Table I, and the thinner plate is actually better rigidized than the thicker plate.

The above discussion applies to infinite plates, and one may inquire if the results will be modified for finite plates. At an infinite plate with a uniform impedance per unit are Z_B the pressure p due to a plane wave incident from the angle ϕ is given by[9]

$$p = \frac{\sin\phi}{\sin\phi + \rho c/Z_B} , \qquad (1)$$

where ρ is the density of water and c the velocity of sound in water. Thus the pressure at the infinite plate becomes zero for $\phi = 0^\circ$ (end fire) for any finite impedance Z_B. The situation is

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[9] V. Mangulis, Radiation of Sound from a Circular Rigid Piston in a Nonrigid Baffle, International J. Engineering Science, Vol. 2, p. 115 (1964).

much better if the nonrigid surface with a finite impedance occupies only a strip in a rigid plane so that only a finite length of a nonrigid baffle has to be traversed before the wave reaches the observation point; for a strip ten wavelengths wide and $Z_B = i\rho c$ the pressure for $\phi = 0^o$ is -10 db on the rigid baffle 1.6 wavelengths beyond the edge of the nonrigid strip, and -6 db 8 wavelengths beyond the edge.[10]

Another indication of the effects of finite patches of nonrigid baffles can be obtained if we consider the reception by an array on an infinite rigid plane baffle. The array elements have a finite impedance Z_I looking into the mechanical terminals from the water side, i.e., a force exerted on the circular piston of an array element is equal to Z_I times the velocity of the piston. If a plane wave is incident on the array at a grazing angle, then the wave will pass over the finite impedance array elements before it reaches the last column, and thus the pressure at the last column will be reduced just as if the plane wave had passed over a finite nonrigid baffle (i.e., the plane wave does not know the difference between nonrigid surfaces which are transducer elements or which are parts of baffle). Consequently, let us examine the force or velocity in the last column of such an array as a function of the length of the array. For an array of 6 rows

[10] V. Mangulis, On the Effects of a Nonrigid Strip in a Baffle on the Propagation of Sound, J. Sound & Vibration, vol. 2, p. 23 (1965).

and N columns, spaced 20 cms apart, circular pistons of radius 6.35 cms, $Z_I = - i\ 5.3\ \rho c$, frequency 3000 cps, the ratio of the maximum velocity in the array to the velocity in the last column is

Table III. A-II

N	Velocity differences, db
12	1.5
24	2.4
36	3.0
48	3.3

given in Table II.[11] Thus the velocity or force decreases as the length of the array is increased. For an infinite array the average pressure on an element near grazing angles is given by Eq. (1) if we replace Z_B by Z_I divided by the ratio of the piston area to the area of the baffle per element,[12] and therefore for an infinite array the pressure approaches zero as $\phi \to 0^o$. Thus again we see that non-rigid baffles of finite extent do not reduce the pressure at grazing angles as much as nonrigid baffles of infinite extent. Consequently we can regard our work on the infinite baffle with a discrete set of resonators[8] as a pessimistic overestimate of the losses suffered due to the finite rigidity of the baffle, and if the infinite plate calculations show negligible losses due to the finite impedance of the plate with resonators, then the losses can be expected to be negligible for the baffle of finite extent also.

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11 A. Kane, unpublished data.

12 V. Mangulis, Infinite Array of Circular Pistons on a Rigid Plane Baffle, J. Acoust. Soc. Am., vol. 34, p. 1558 (1962).

In summary, the calculations show that resonators or stiffeners spaced at distances of 4 cms (or less) rigidize a 1/4 or 1/2 inch thick plate sufficiently so that the pressure levels are acceptable over a bandwidth of 1000 cps.

One should note that in an actual baffle design the spacings between resonators will probably be irregular, in which case the pressure levels may be improved, because a regular spacing (which was assumed in the above calculations) permits the development of flexural waves in the plate and supports various resonance effects which may be suppressed by an irregular spacing.

One might think that the performance of the plate could be just as well improved by adding the mass of the resonators to the plate, thus making the plate thicker. However, an increase in the thickness of the plate also increases the flexural wavelength, and if the wavelength in water divided by $\cos\emptyset$ matches the flexural wavelength, the pressure at the plate is zero. For example, if we had added the resonator mass at 4 cm spacing to the 1/2 inch plate, then at 2500 cps the wavelength in water would match the flexural wavelength in the plate at end-fire, and the thick plate would act as a pressure release surface near end-fire.

B. Experimental Results

1. General

The baffle, the structure which houses the flush transducers, has been investigated analytically for the impedance that it exhibits to incident acoustic energy. The prime utility of the baffle is maintaining a high impedance to both normal and grazing waves.

This same baffle, however, as part of a ship must maintain seaworthiness of the vessel. This would be no problem if the baffle could be of conventional design, i.e. framing on rather small centers for both the horizontal and vertical directions, with plating of moderate thickness to carry the hydrostatic pressure loads and skin stresses. The design of a baffle for high impedance through the sonar band of interest demands large panel sizes, a specific configuration for the sea chests that cover the transducers, and resonators between the sea chests.

The baffle with transducers must withstand exposure to battle situations at the worst, and all the design conditions for a sea going vessel. Structural loadings on the panel result from ship motions and the hydrostatic pressures of immersion. A very intense structural loading to the baffle occurs in the presence of a nearby underwater explosion. This situation is a design condition for the array. Closely related to the structural problem is the noise isolation problem. A

compromise always exists in the degree to which the baffle should be coupled to the ship's hull. A stiff coupling solves structural problems and allows a good path for ship's self-noise to the baffle. A more flexible coupling is inferior structurally, but reduces the noise level at the transducers.

These topics were discussed in TRG-023-TM-66-7 entitled "Structural Considerations in Detection of Array Configurations" by I.Melnick. The subject material is reproduced below (para. III.B.2) because it is pertinent to the problem of evaluating approaches to designing a baffle.

Scale model impedance tests have provided a rapid and inexpensive way of evaluating experimentally, a series of hull configurations. Although the test information cannot be as valuable as that obtained from a full scale reproduction of the baffle in water, trends and comparisons can be relied upon.

The test data of paragraph III.B.3 is identical to Report No. TRG-023-TM-67-1 entitled "Scale Model Baffle Impedance Tests" (U) by I.Melnick. This material is added here to present a fuller picture of the theoretical and experimental evidence for the TRG baffle concept in a single document.

The design of the TRG baffle leads to welding problems that are more sever than those associated with normal ship's framing. Previous analyses and experimentation has shown that a solution to these welding problems is available in a state-of-the-art

electron beam welding system. The section entitled "Position on Electron Beam Welding" supplies the history and details of the baffle welding problem.

A section of this position paper describes the TRG position on an array-covering membrane. Although this subject is not of immediate significance to the project, the subject matter is related to baffle design. The particular paragraph (III.B.5) is entitled "TRG Position on the Array-Covering Membrane."

2. Structural Considerations in Selection of Array Configurations

A sonar array utilizing an air-backed, rigid, baffle generates specific structural requirements. Some of these requirements are severe enough to initiate investigations into novel structural designs for hull plating.

A description of the loading conditions is a logical starting point for an investigation into structural problems.

The sonar array must be located below the water surface to effect an efficient transfer of acoustic energy. It also should be placed as deep as possible to prevent cavitation in transmission, and surface noise and bubble effects in reception. Lowering of the physical location of the array increases the hydrostatic forces acting on its outer surface, and results in a high external pressure even with the ship stationary in still water.

A ship is exposed to increased hydrostatic forces on its plating when it moves in a heavy sea. The forward portion of the ship, particularly, is exposed to slamming loads which can amount to approximately three times the static pressure forces. Conventional ship's framing is adequate to withstand these forces with economical utilization of small panels and numerous frames and longitudinals.

Combat ships have an additional hydrostatic force which must be resisted with a limited damage level. This force results from a nearby underwater explosion. The ship senses the rapid pressure rise of the shock wave from the explosion. Surface ships also experience a rather large upward acceleration immediately after the initial sharp pressure pulse passes. Again, as in the case of hydrostatic loading, conventional

ship design yields an economical arrangement of plating, frames and longitudinals to resist underwater explosions to an adequate degree.

The location of the array as far underwater as possible places the array structures far from the centroid of the ship's bending cross-section. This location induces large tension or compression stresses in the plane of the hull from the ship's bending moments in the vertical plane. Compressive stresses acting in the plane of a hull panel can induce instability, especially when the lateral forces of the hydrostatic load are also acting. This instability can induce failures at weak spots like welds and at cutouts in panels where transducer sea chests replace hull material. As with the previously described loads, conventional ship framing with small panel dimensions is not as sensitive to panel buckling as the anticipated sonar plating.

A description of the sonar hull plating is helpful in justifying the above statements. The spacing of the transducers of an array should be determined by acoustic requirements. Compromises with a uniform spacing of transducers must be made for the vertical and horizontal members that are required for the ship's structural integrity. In addition, the uniformity of the transducer array is disturbed by the fact that at least one vertical butt weld is required every 8 feet for manufacturing practicability and this weld cannot be near the ship's vertical structural members.

Every structural member added after considering the above mentioned items will further translate the transducers from their optimum location and lower the array's transmission and reception efficiency. Additional structural members also reduce the effectiveness of resonators unless the members act as extremely deep beams. In the sonar keel, for example, bulkheads which are solid except for a reinforced access hatch do not lower hull impedance, but normally sized structural tees impair the high impedance attained by resonators. Thus, the largest sized panels that meet all structural requirements

will yield the most satisfactory overall sonar performance. The distance between major frames is a practical limit for the longitudinal extent of the typical panel. This dimension is 7 feet for "Fletcher" class destroyers and is 8 feet for a ship like USS SPOKANE. The vertical dimensions of the panel is determined by the desired height of the array, and is limited by structural requirements.

There are beamformer designs under development which can easily compensate for any radius of curvature of the hull plating or any deviation of the array pattern from theoretical. In the event that such a beamformer is used in the C/P array, the hull shape will be determined only by a naval architectural decision and Section B can be disregarded.

Intermediate structural members, if introduced, limit the number of resonators that can be placed between sea chests which further detracts from the attainable plate impedance.

In summary therefore, environmental loading of ship's hull plating becomes a problem when the panel size is increased as presently intended for the C/P array. At worst, a compromise will have to be made between sonar performance and structural integrity. At best, panel sizes will be large enough to insure adequate sonar performance and structural integrity will be established both analytically and experimentally.

Particular aspects of the problem are discussed in the following paragraphs.

a. Influence of Hull Radius of Curvature on Gain of an Array

An array of transducers arranged in a cylindrical surface will suffer a loss in gain as compared to that same array located in a flat surface, if the curvature is not compensated for. An attempt to eliminate the loss in gain will result in a more complicated, more costly, and less reliable beamformer. An indication of the effect of radius of curvature on gain of the array is tabulated in Figure III.B-1. The calculations were

HULL RADIUS OF CURVATURE IN FEET	ONE WAY GAIN LOSS IN DB
12	2.5
16	1.4
20	0.9
24	0.6

FIGURE III.B-1 GAIN LOSS AS A FUNCTION OF RADIUS OF CURVATURE

made for an array of 12 elements in height at a frequency of 3 Khz and the losses are given for the worst steering angle.

b. Resistance of the Hull Plating (Formed Convex Outward) to Hydrostatic Pressure

The choice of configuration for the hull plating of a curved panel with the convex side facing the sea is based upon the high rigidity and low stress levels that result from the hydrostatic loading of that configuration.

A cylinder or portion of a cylinder resists external or internal hydrostatic pressure by inducing membrane stresses in the circumferential and axial direction. The stresses in the axial direction from hydrostatic forces are small in shipboard problems because of the large cross sectional area of the ship available to resist these loads. The magnitude of circumferential stresses are a function of plate thickness, external pressure and the radius of curvature. The radius of curvature should be made small to limit the circumferential stress, but this is also a parameter of the sonar design as described above. A flat plate is acoustically desirable but would suffer from excessive stresses and deflections under the hydrostatic pressure. A compromise is required to optimize the performance of both the sonar and the structure. From the structural point of view, the required thickness of hull plating as a function of plate curvature is shown in Figure II.B-2. The criterion for establishing the minimum required thickness is buckling from external pressure. A design load of 34 psi was used, that being a rather severe level. The conclusions to be drawn from Figure 2 is that reasonable hull thicknesses can be relied upon to design a hull free of buckling limitations.

c. Resistance of a Concave Hull of Varying Radius of Curvature to Hydrostatic Pressure

If the hull plating is formed to a concave outward radius of curvature, the buckling problem of the convex outward configuration is eliminated. However, substituted for

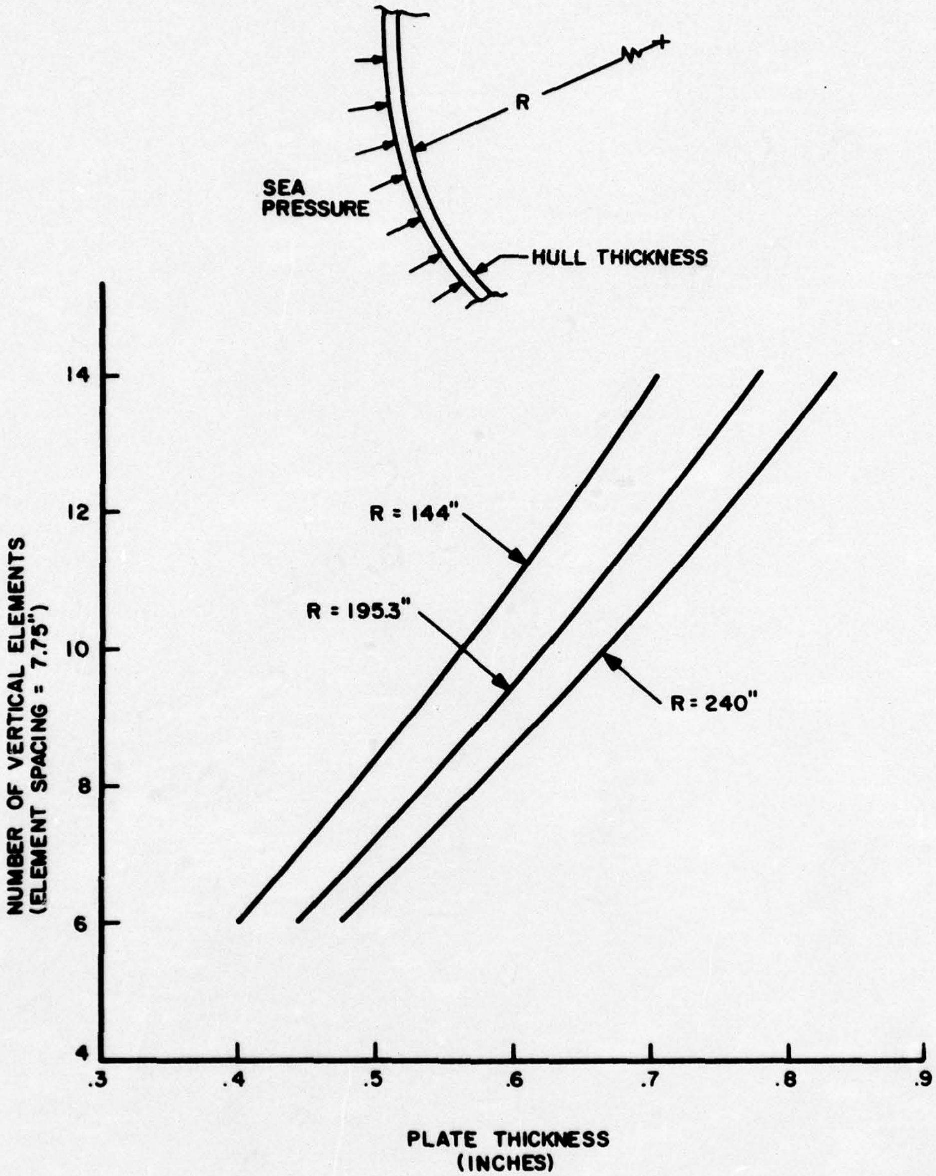

FIGURE III.B-2. MINIMUM PLATE THICKNESS TO ENSURE ELASTIC STABILITY OF A CURVED PLATE OF RADIUS R UNDER A DESIGN PRESSURE OF 34 psi. (End Conditions considered as Half-Way Between Hinged and Fixed).

the buckling problems is the fact that tensile diaphragm stresses now appear rather than compressive stresses. Tensile stresses ara far more serious because tensile failures are more dangerous. This is of even greater consequence when a large amount of welding is present, as in the present case. A saving feature arises in that the welding technique under consideration, electron beam welding, has shown excellent strength values, and high consistency.

The required thickness for a varying plate radius of curvature under the concave outward configuration is shown in Figure III.B-3. Again, we can conclude that reasonable plate thicknesses will withstand the loading.

d. <u>Array Plating Resistance to Shock Loading</u>

Shock loading of the array plating will result in the maximum stress levels seen by the plating. The analysis of a plate under shock loading indicates that the plate dimensions and the thickness of the plate are the most sensitive parameters. The radius of curvature of the plating will affect the stress level to a lesser extent.

One complicating feature of the shock loading problem is the determination of the allowable stress level, or deformation, to be accepted within the confines of resisting an underwater explosion and being able to continue combat and return to base.

Even when allowable stress levels are established, the analytical solution to the problem of determining an acceptable plate thickness is complicated by the fact that plastic deformation is involved.

The program for establishing the resistance of the sonar plating to the design shock condition must then make use of adequate experimentation, in addition to analysis, to assure a reliable structure.

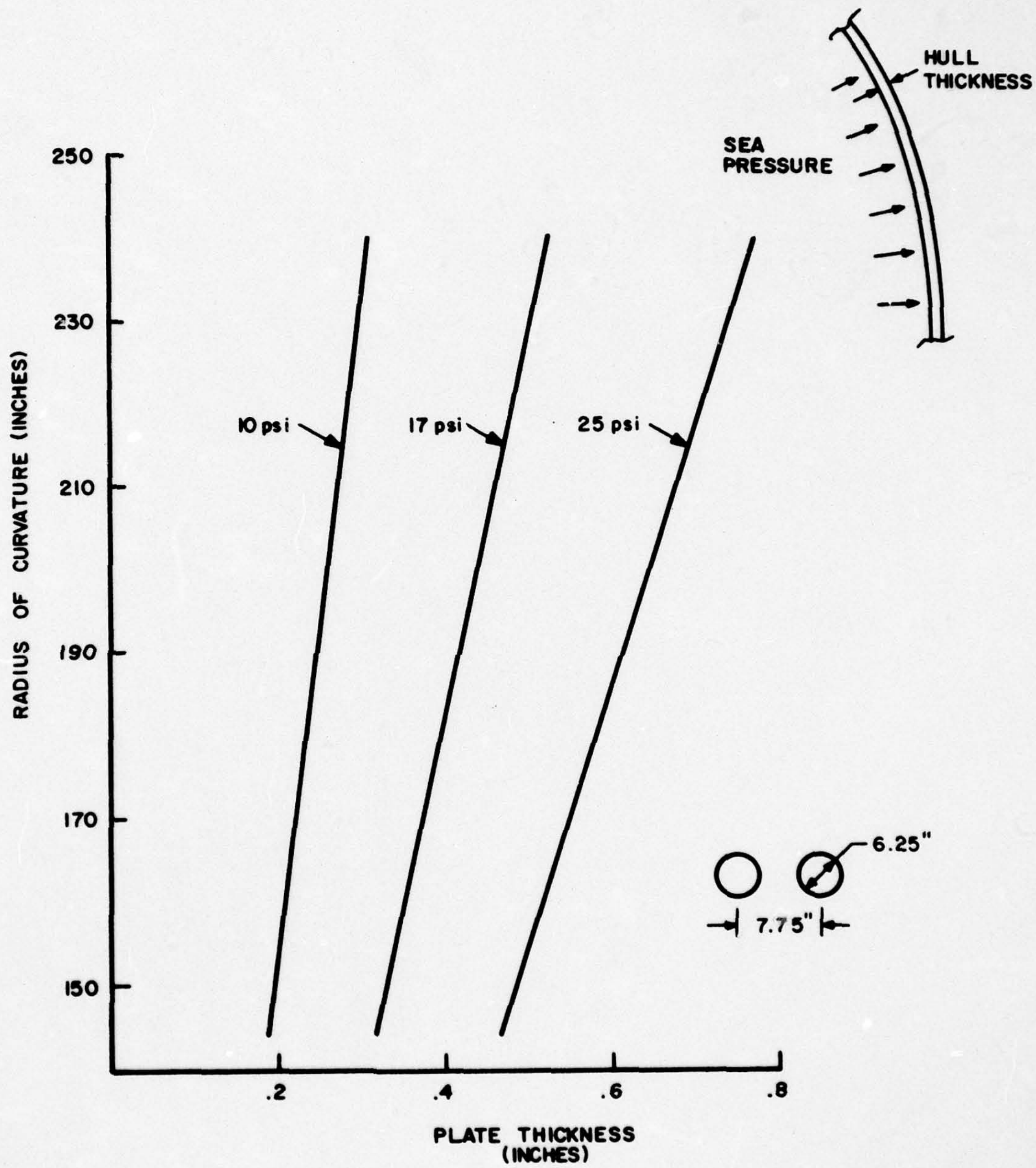

FIGURE III.B-3. MINIMUM PLATE THICKNESS TO INSURE A MAXIMUM DIAPHRAGM STRESS OF 40,000 psi.

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TRG'S POSITION PAPER ON THE CONFORMAL/PLANAR ARRAY DESIGN.(U)
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It is too early in the program to establish the panel sizes for various plating thicknesses, but the restraint does exist. One can reasonably state that a concave outwardly shaped hull with good welding will be safer than a convex outwardly shaped hull because the danger of buckling is eliminated.

e. Effects of Hull Shape on Sea Chest Geometry

The hull shape influences the sea chest geometry. A flat hull allows the inboard end of the sea chest a definite maximum dimension for the bolting arrangement.

When a convex outward hull form is chosen, the maximum dimension of the sea chest end is limited by a decrease in the radius of curvature and is always smaller than for a flat hull.

When a concave outward hull form is utilized, the maximum dimension for the sea chest end is always greater than for a flat hull and increases with decreasing radius of curvature.

It is important to design the inboard end of the sea chest with a reliable joint that eliminates leakage from the sea.

f. Isolation From Ship's Noise

It may be of advantage to the overall system performance to isolate the sonar structure from the self-noise sources within the ship. This will result in a higher signal-to-noise ratio and attendant advantages.

When sonar hull plating is incorporated in a "blister", isolation takes the form of a peripheral joint about each section of hull plating. A "soft" mount for the plate will reduce the noise levels in the plate. Naturally, the plate's resistance to hydrostatic and shock forces will be lowered by the mount. This lowering of structural strength is strongly dependent upon the plate dimensions.

When the sonar hull plating is incorporated in a sonar keel, it seems likely that the isolation joint can be placed between the sonar keel and the ship. In this configuration, the maximum sized sonar panel is not limited by requirements for isolation.

g. Alternative Solutions to the Excessive Structural Loading of Large Panels.

In the event that large panel sizes are not safe from the structural viewpoint, a compromise may be made in the introduction of intermediate stiffeners and the acceptance of a reduced sonar efficiency.

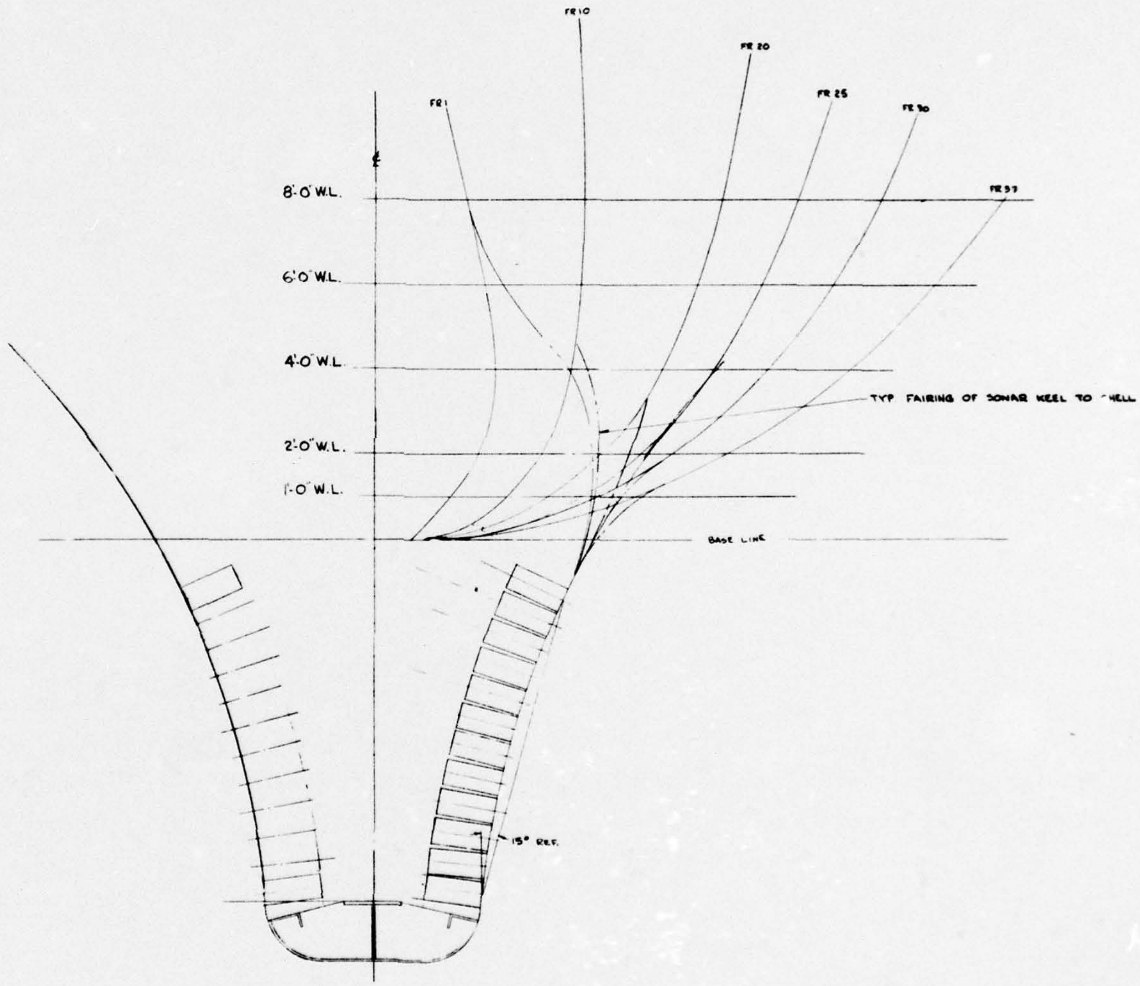
Another possible solution to the problem of inadequate structural strength for a large panel is the introduction of an intermediate stiffener which comes into play only when large plate deformations occur. This stiffener will traverse the plate but not touch it. A definite air gap at one or more points will be maintained between the plate and stiffener. When large loads appear on the plate and large deformations result, the air gap will be eliminated and the intermediate stiffeners will be loaded. This system offers the advantage of the strength of the intermediate stiffener without compromising the acoustic impedance.

h. Sonar Keel Proposal Drawings

The drawings included in this report, SK 023-114 and SK 023-115, show how an array will be fitted to the contour of existing ships. In both cases, an array of 12 elements in height is shown inclined at an angle of 15° to the vertical.

Both arrangements require approximately the same additional height added to the ship, but the concave hull shape needs the extra 7 inches to accommodate transducer replacement on row 1.

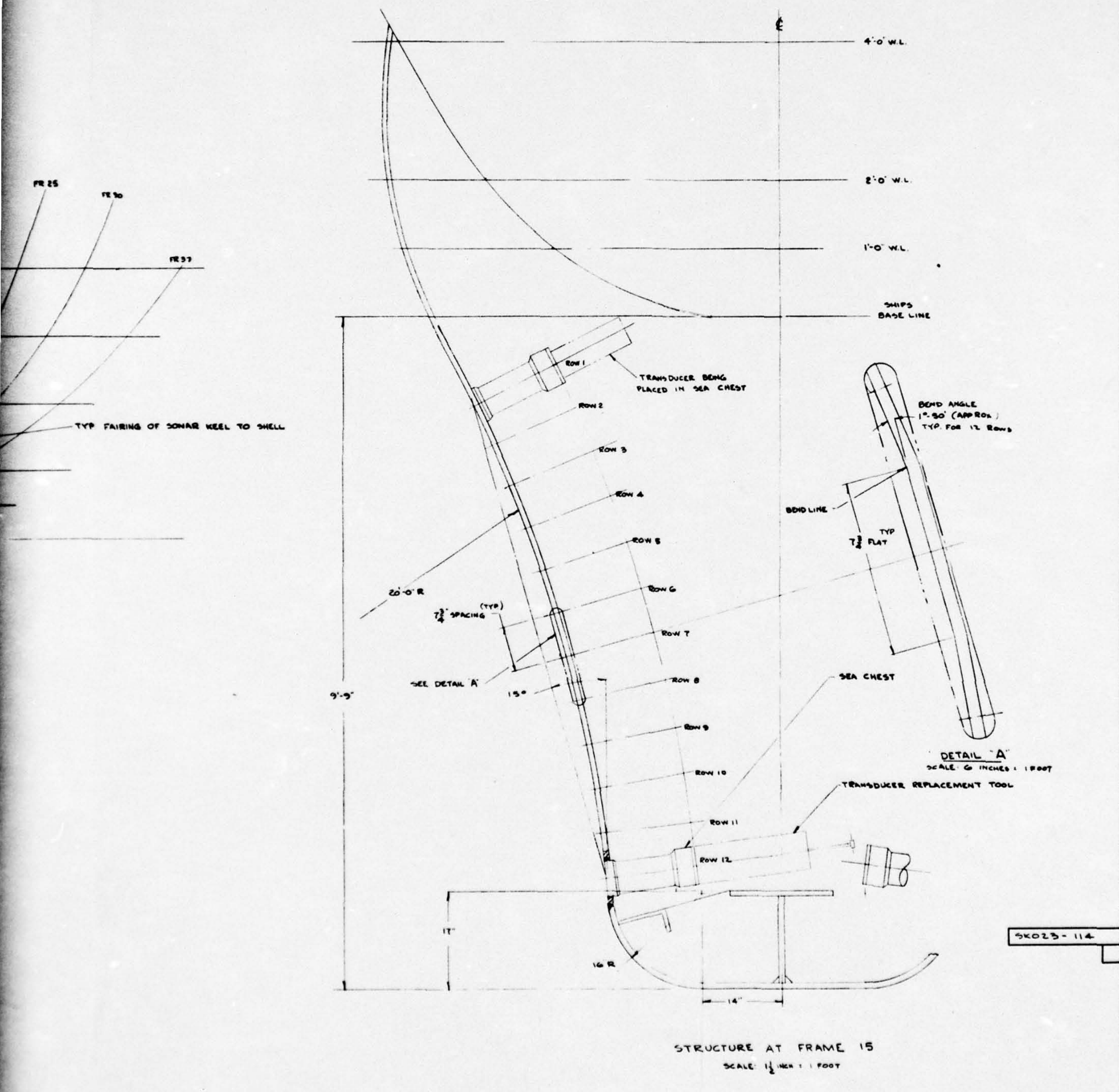
Both designs can be faired into the ship's lines with the concave design showing a smooth transition.



GENERAL ARRG'T OF STRUCTURE

SCALE : $\frac{1}{4}$ INCH = 1 FOOT

REVISIONS				
SYM	DESCRIPTION	ECO NO	DATE	APPROVAL

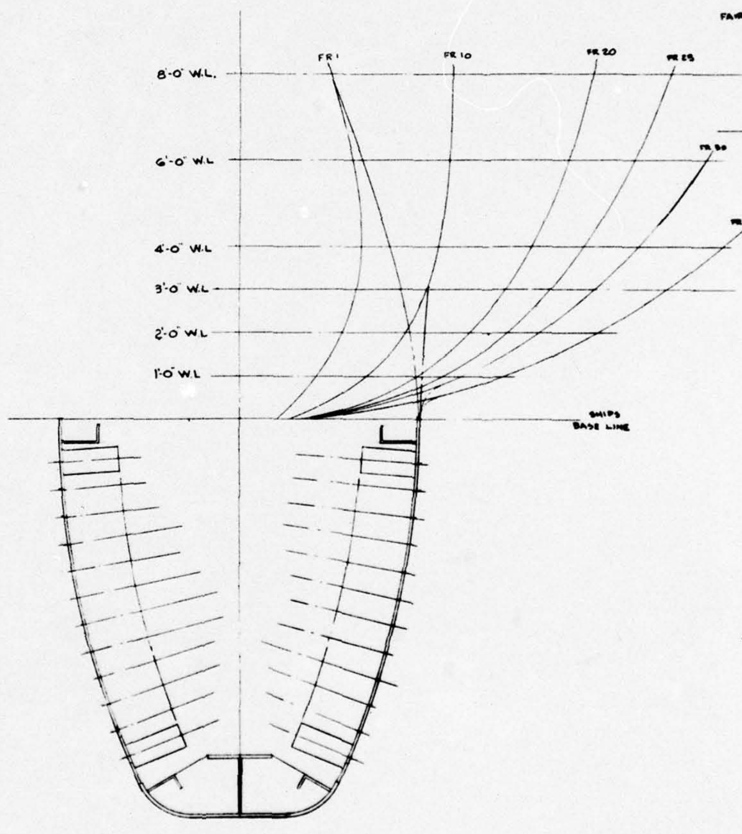


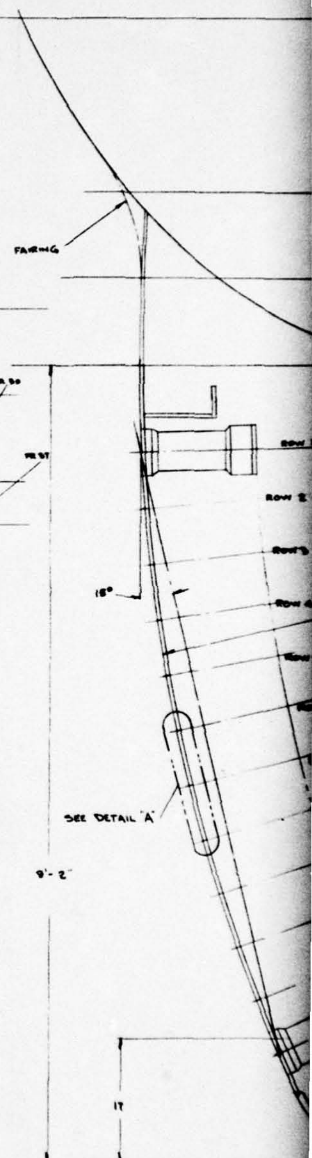
STRUCTURE AT FRAME 15

SCALE: 1½ INCH : 1 FOOT

ITEM	QTY REQD	PART NUMBER	DESCRIPTION	MATERIAL	SPECIFICATION	NOTE
LIST OF MATERIALS OR PARTS LIST						

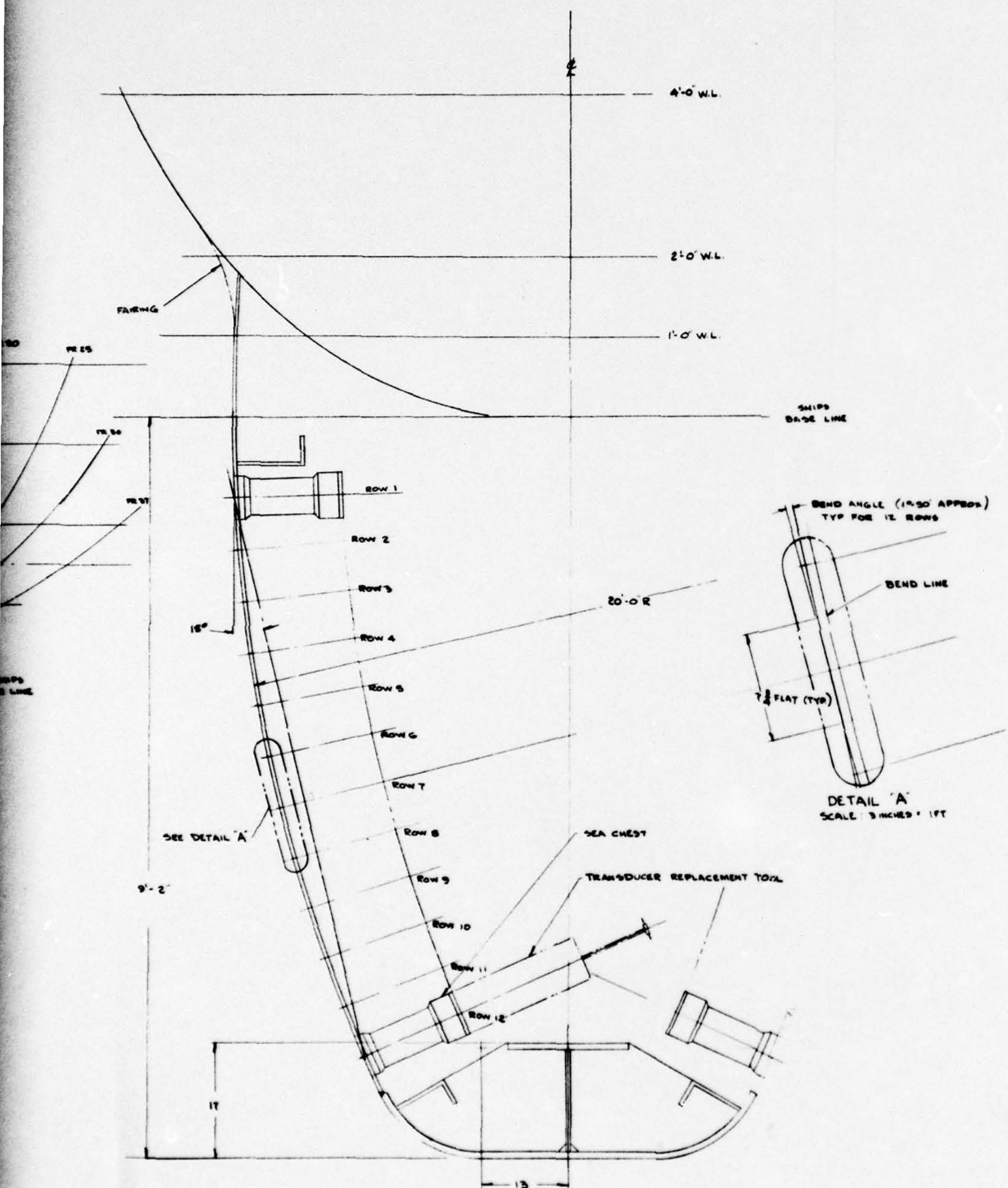
UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES TOLERANCES ARE .XX ± .03 .XXX ± .005 .XXXX ± .0005 ° ± 0.5°	CONTRACT NUMBER Nobsr 93203	ROUTE 110 MELVILLE, NEW YORK
MATERIAL	DRAWN BY J.R. DATE	ARRAY STRUCTURE PROPOSAL 20FT CONCAVE RAD, 15° TILT, 12 ROWS
FINISH	CHECKED BY	
	DESIGNED BY	
	APPROVED	
	DESIGN ACTIVITY APPROVAL	CODE IDENT. NO. 14059 SIZE E NUMBER SK023-114 REV





GENERAL ARR'GT OF STRUCTURE
SCALE $\frac{3}{4}$ INCH = 1 FOOT

SYM	DESCRIPTION	ECO NO	DATE	APPROVAL



DETAIL 'A'
SCALE: 3 INCHES = 1 FT

STRUCTURE AT FRAME 15
SCALE: 1 1/2 INCH = 1 FOOT

SK023-115

ITEM	QTY REQD	PART NUMBER	DESCRIPTION	MATERIAL	SPECIFICATION	NOTE

LIST OF MATERIALS OR PARTS LIST

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES TOLERANCES ARE .XX ± .02 .XXX ± .005 .XXXX ± .0005 X° ± 0.5°	CONTRACT NUMBER Nobsr 93023	ROUTE 110 MELVILLE, NEW YORK
MATERIAL	DRAWN BY	DATE 66
FINISH	CHECKED BY / DESIGNED BY / APPROVED	ARRAY STRUCTURE PROPOSAL 20 FT. CONVEX RAD., 15° TILT, 12 ROWS
	CODE IDENT. NO. 14059	SIZE E NUMBER SK023-115 REV

2

3. Scale Model Baffle Impedance Tests

The accompanying photographic prints were reproduced from the acceleration records made of the response of various acoustic baffle models that were tested in the TRG Acoustic Test Chamber.

The initial object of the experiment was to obtain experimental data on the mechanical impedance of baffles proposed by TRG and G/D. The acoustic test chamber had already been used to obtain data in air which agreed with the experimentally determined behavior of the same baffle configuration in water. An analysis of scaling laws suggested that we could scale frequency and physical dimensions inversely.

The TRG proposed hull configuration consists of plating with sea chests and resonators.

Two sets of data for the G/D baffle are presented: a slotted aluminum block with cylindrical holes for transducers of 4-inch and 5-inch diameters.

Sets of data for 1/6 scale model tests and 1/4 scale model tests are presented. Historically, the 1/6 scale models were chosen first due to the ready availability of raw aluminum block of the necessary thickness for the G/D model. Later, thicker sections were found to be available and a 1/4 scale model was made. The availability of data from both scales allows further substantiation of the scaling laws being used.

The reference for the experiments is a steel plate of 3/4 inch full scale thickness, this being representative of existing hull plating near the keel of a large ship.

The data recorded in this experiment is the output voltage of an accelerometer mounted on the particular test piece. The test piece is accelerated by an acoustic wave

produced by a loud speaker. The accelerometer is mounted on the side of the test piece that is not insonified. A special effort is made to prevent the acoustic field from leaking around the joint between the test piece and the acoustic test chamber to minimize the pressure response of the accelerometer, which was found to be a problem. Accelerometers were first mounted on the insonified side to provide a greater number of possible accelerometer locations on the TRG models where the other surface was covered by sea chests and resonators. The requirements for reliable data required that freedom in the location of accelerometers be compromised. The solution was an accelerometer located on the non-insonified side of the brass plug which represented the transducer in the model. The two locations where accelerometer data were taken were prepared by cementing the simulated transducers to the hull structure. All other simulated transducers were isolated from the hull structure.

Data were taken on a point at the center of the model and near an edge to determine the uniformity of the plate response.

The loud speaker location within the test chamber was moved to give head-on insonification (normal incidence) and insonification at 45° to the plate (grazing incidence).

a. Data Analysis

Figure III.B-4

The lower half of this plate shows the geometry of the test samples. The upper graph illustrates the effect of the pressure sensitivity of the accelerometer when the accelerometer was mounted in the test chamber (in acoustic field) and outside the test chamber (without acoustic field).

The lower graph is a check of accelerometer mounting technique. The red and green curves compare the response of the accelerometer when screwed in plate (a result of locating the accelerometer on the simulated transducer) with the response when cemented in plate (the usual procedure when mounting accelerometers on the insonified surface). The green and blue curves compare the response of two different accelerometers sequentially taken at the same location.

Figure III.B-5

In this plate the acceleration response of the TRG hull model is compared to the response of a reference plate for the frequency band between 9 and 19 kc (full scale frequency range 1.5 to 3.16 kc). The length of the model's sea chests is scaled to a full size length of 16 inches.

At grazing angles of incidence, a considerable reduction in plate acceleration is accomplished throughout the active sonar band, and below it.

A smaller effect is noticed in the data taken at normal incidence. Fortunately, a high hull impedance is not as necessary at this angle of incidence.

It is reasonable to expect that even lower accelerations will be recorded with larger sized test plates. This model is only one quarter of the size of the panel being proposed for the array.

Figure III.B-6

This data compares the reference plate, the GD model with 5" holes (full scale), and the TRG model with 19 inch sea chests (full scale). Comparing the response of the 16 inch long sea chests from Plate 2, the longer sea chests result in a lowered impedance at the higher frequencies of the sonar band.

The data on the GD model suggests there are many modes of vibration throughout the sonar band. These 1/6 scale test results are worse in this respect than the 1/4 scale tests described later.

Figure III.B-7

This 1/6 scale model test was performed on a block of aluminum slotted into the shape shown. Although it would seem that this plate would show a high impedance because of the large number of "resonators," the response was only slightly better than the reference plate. The mass, however, of the slotted block is 10 times that of the reference plate. It should then show a response 20 db below the reference plate. This is accomplished at only a few frequencies. A possible explanation for this effect is overcoupling through the basic "plate" between resonators.

Figure III.B-8

Plate 5 shows the response of 1/4 scale models of the GD hull form with 4 inch diameter holes compared to a reference plate. The low frequency response is good, but the effectiveness of the design is limited to 11 kc on the scale model (2.75 kc, full size).

The benefit of changing the hole size from 5 inches (Plate 6) to 4 inches is evident. The frequency where the stiffening effect is lost was moved from 9.5 kc (2.37 kc full size) to 11 kc (2.75 kc, full size).

Figure III.B-9

This plate shows the response of the 1/4 scale GD hull model with 5 inch holes compared to the reference plate. As previously mentioned in the description of Figure 4, the plate response above 9.5 kc (2.37 kc full size) was inadequate.

Comparison of this plate with Figure 3 shows the effect of changes in scale.

The prominent peak in the acceleration response at 14.4 kc in the 1/6 scale (2.4 kc full scale) in Figure 3 corresponds well with the prominent peak at 9.7 kc in the 1/4 scale (2.425 kc full scale) in Figure 6. However, the 1/6 scale model shows excessive acceleration below 14.4 kc as compared with the 1/4 scale. This suggests that the smaller the scale the less dependable, and probably the more pessimistic the result.

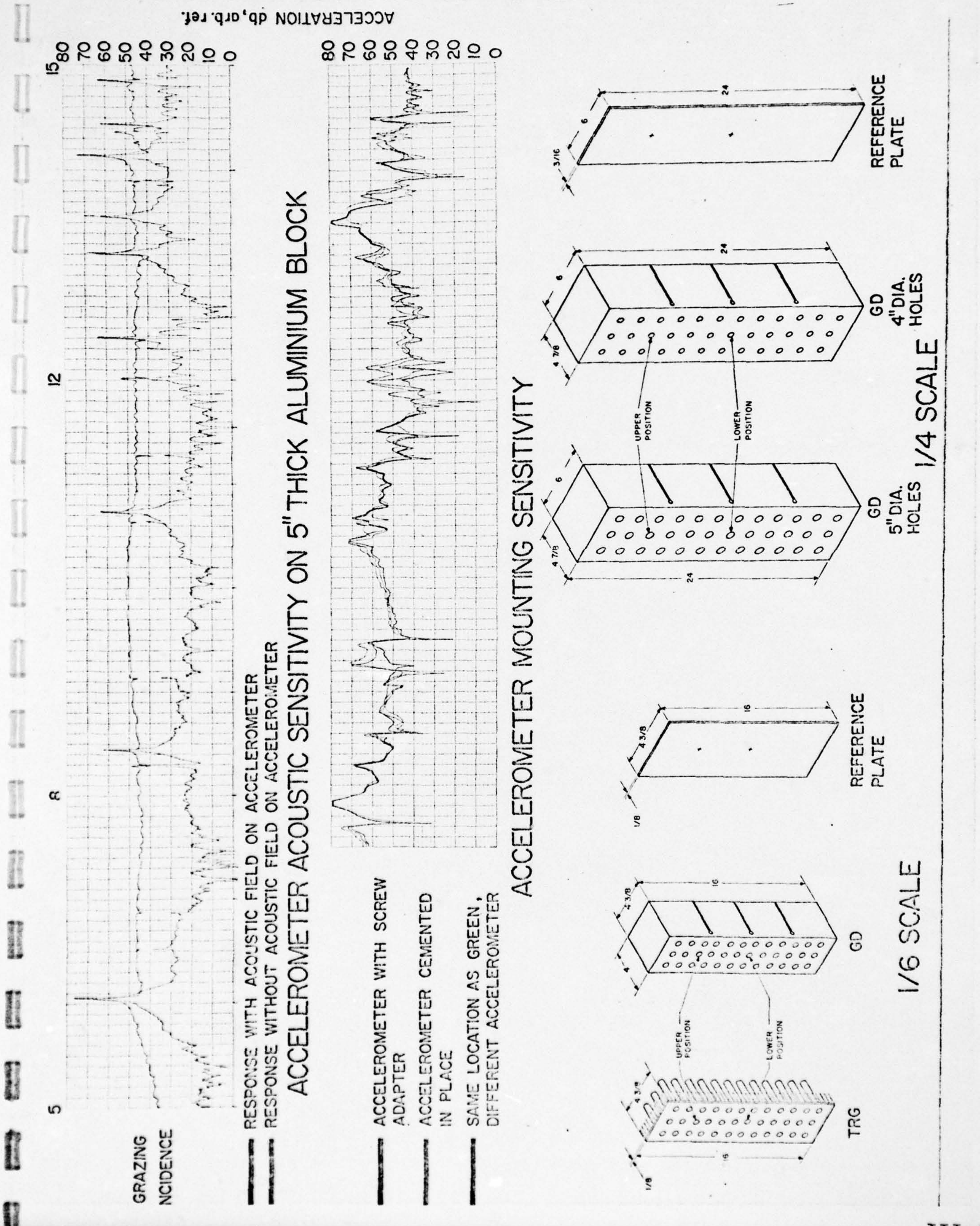
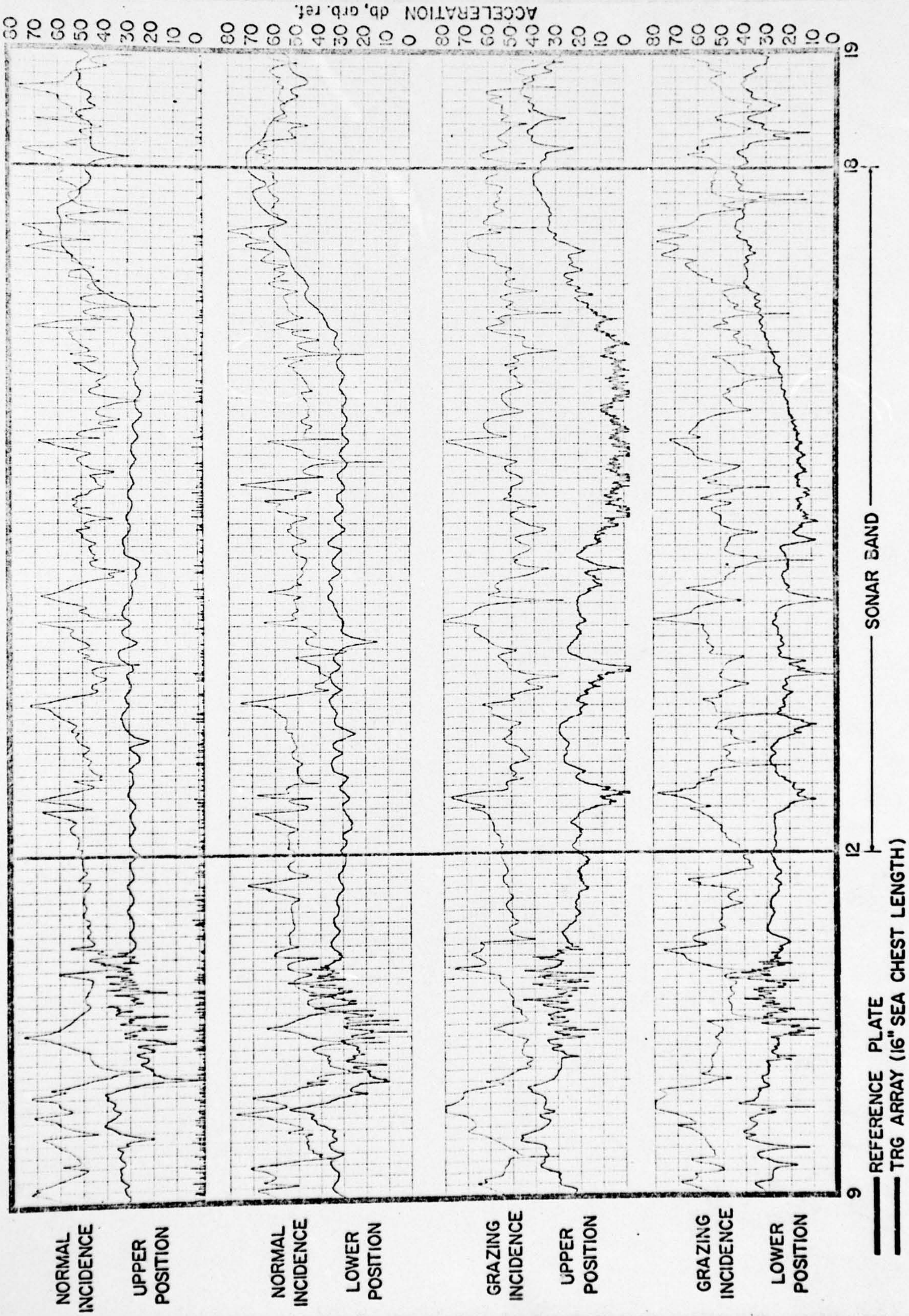
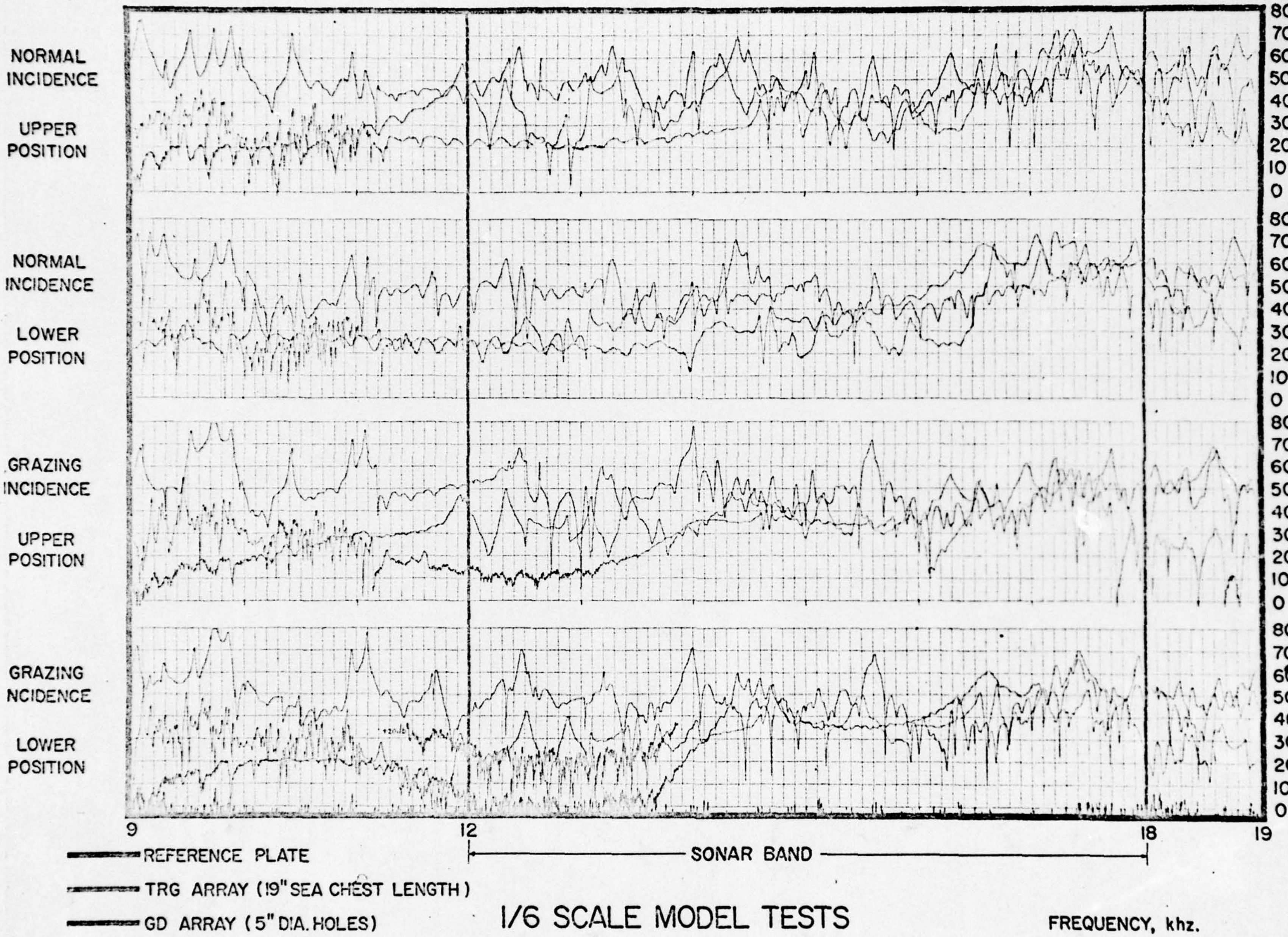


FIGURE III.B-4



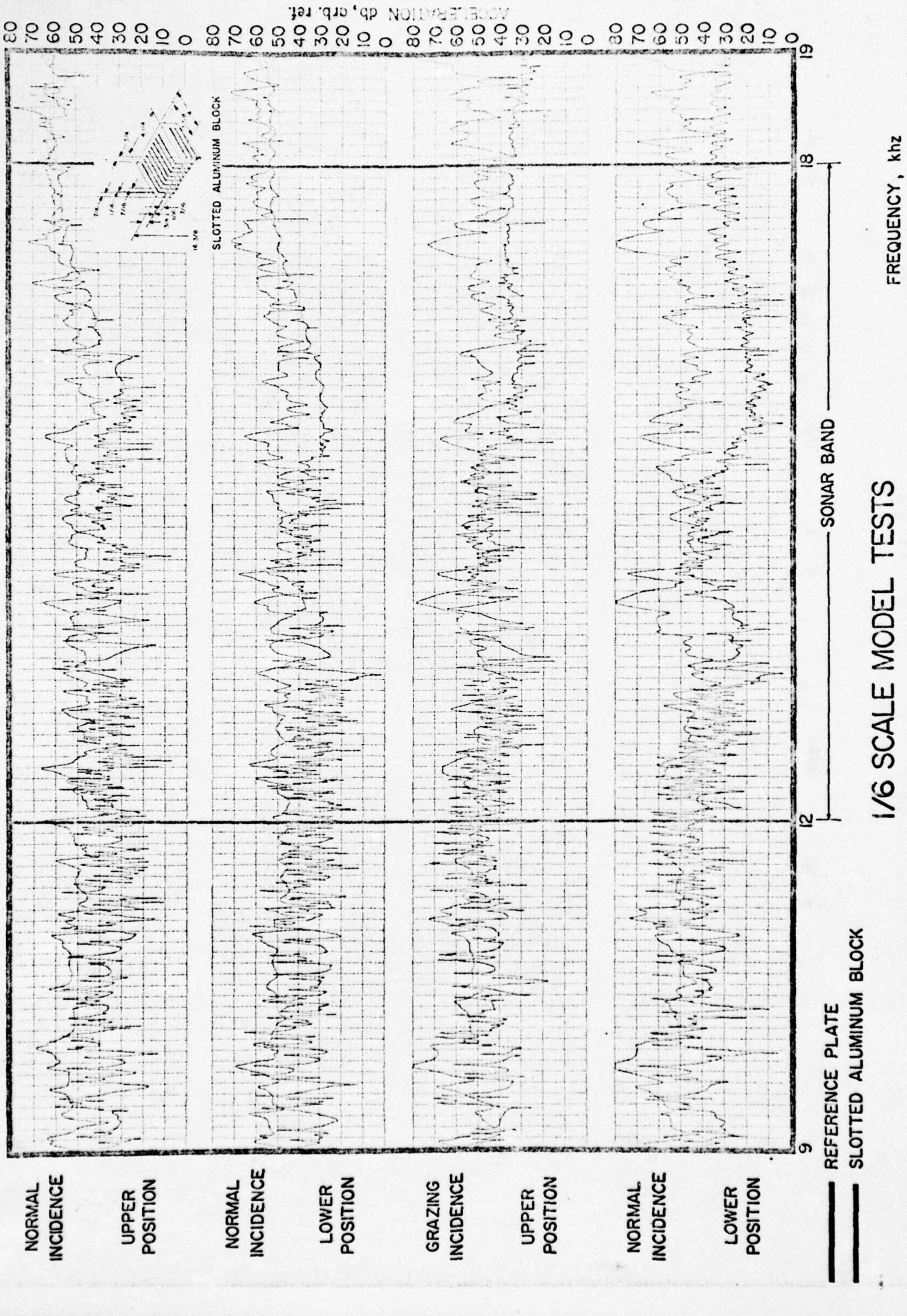
1/6 SCALE MODEL TESTS

FIGURE III.B-5



1/6 SCALE MODEL TESTS

FIGURE III.B-6



1/6 SCALE MODEL TESTS

FIGURE III.B-7

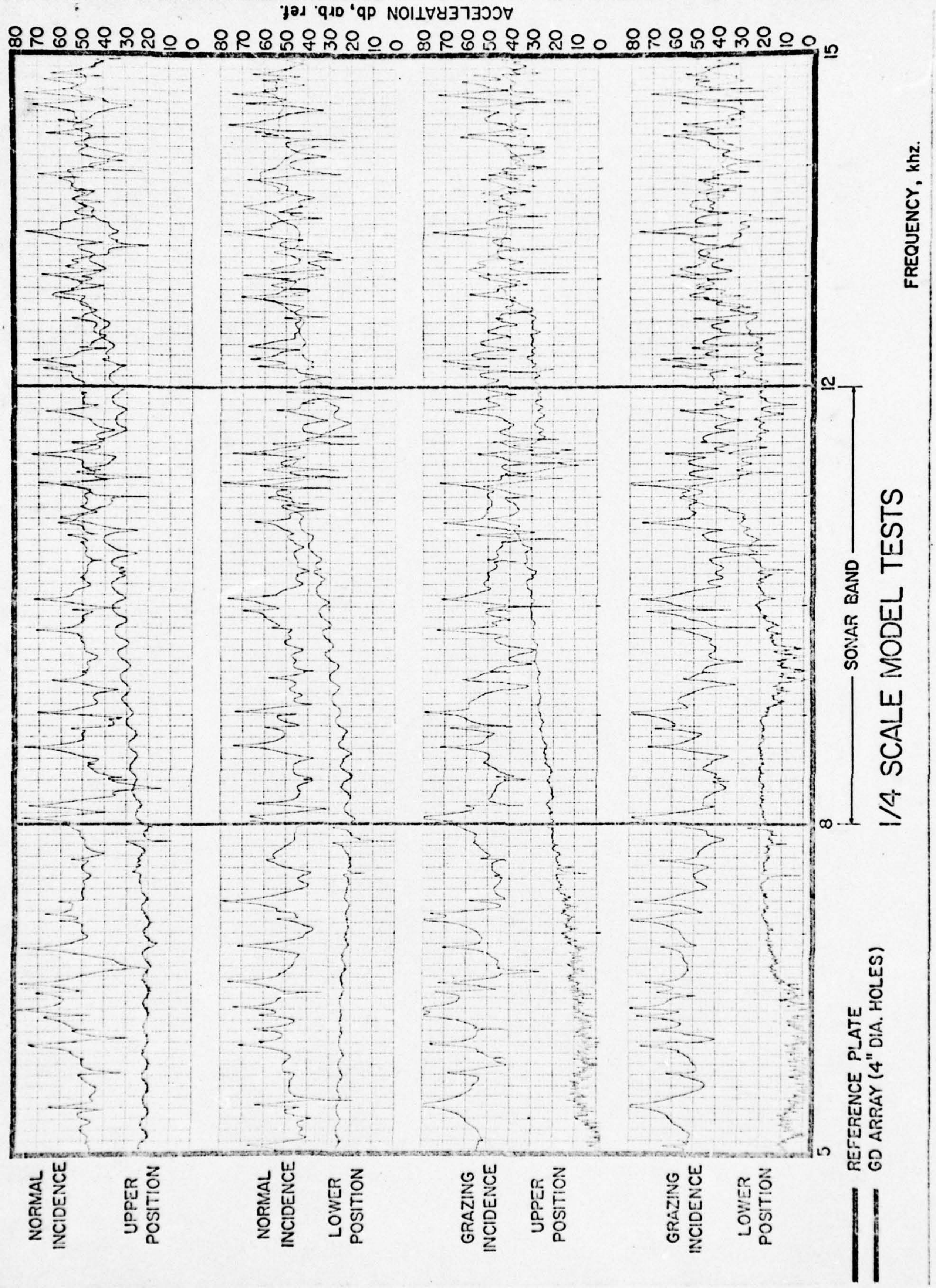
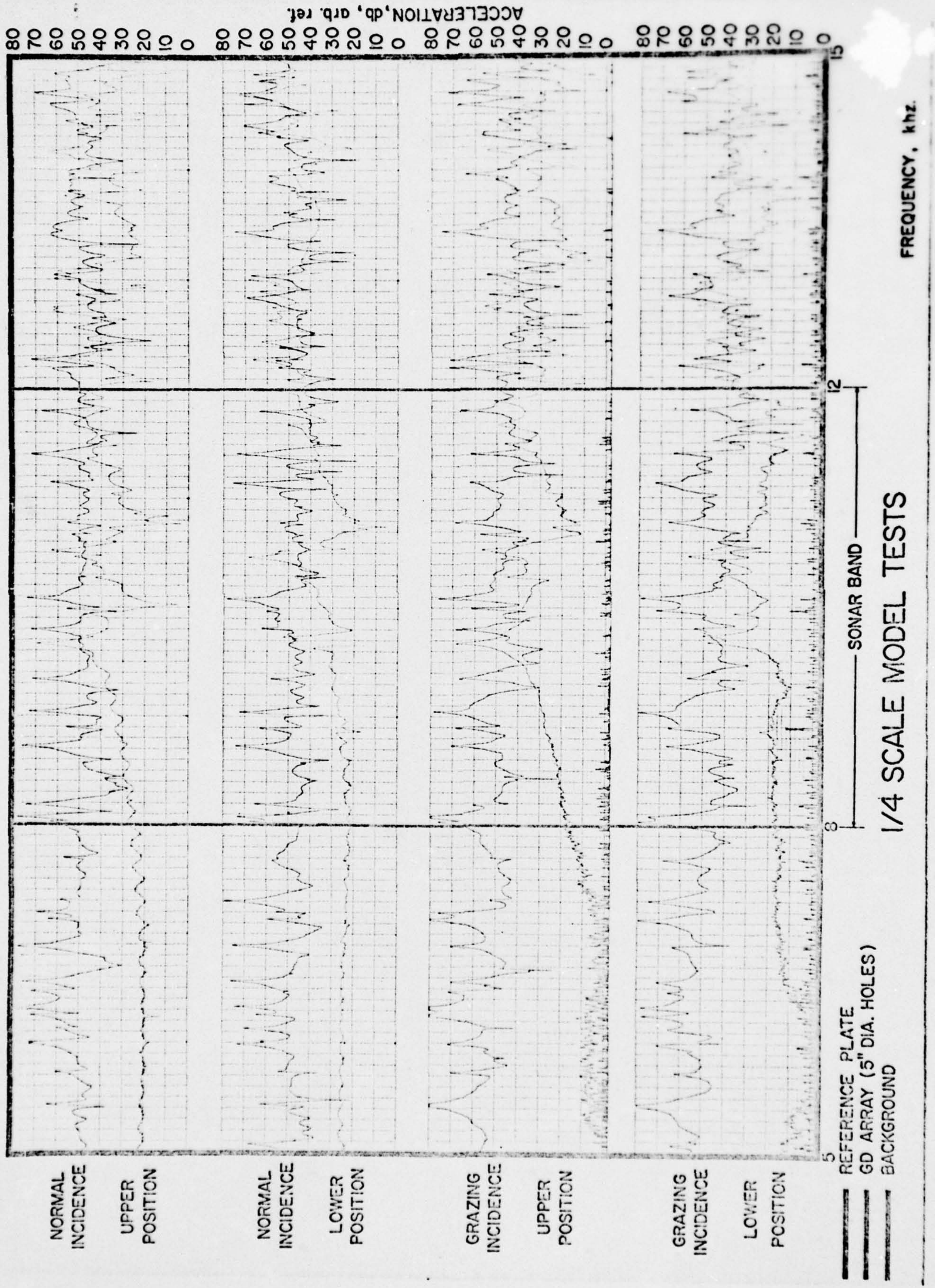


FIGURE III.B-8

CONFIDENTIAL



1/4 SCALE MODEL TESTS

FIGURE III.B-9

CONFIDENTIAL

4. Position on Electron Beam Welding

The technique used by TRG to establish an adequate mechanical impedance for the baffle requires that attention be paid to how such a baffle can be constructed.

In simplest terms, the baffle consists of plating with holes at regular intervals. The holes are filled by attaching tube like sea chests and welding an external lip on these sea chests to the plating. In addition, between holes, resonant stiffeners are added, again by welding.

At first glance, the fabrication of such a structure does not present any special difficulties. Investigation of the requirements of the design reveals the complications. Some important facets of the problem are:

1. At first cut, rather large panels are under consideration. Hull sections eight feet high and eight feet long with 144 sea chests per panel are being investigated because this arrangement is efficient from the array viewpoint. These large panels now have no internal stiffeners to maintain the plating shape, at present a section of a large radius cylinder. The welding of the 144 sea chests must be accomplished entirely from the side opposite from the sea chests because of tight sea chest spacing. Conventional welding will leave a weld cross-section which is wider at the side from which the welding is accomplished as compared to the other side, at least for plates of the thickness being considered. A weld with such a cross-section will shrink more upon cooling

on the welder's side and the plate will gradually form into the sector of a sphere from this deformation. An attempt at removing this deformation can be made by peening after each weld pass, but results are never satisfactory in terms of straightness of the final weldment. TRG's experience with this problem dates from the fabrication of the Dodge Pond Test Vehicle. At that time, all attempts at removing distortion such as peening, staggered welding and even annealing for 24 hours at stress relieving temperatures could not eliminate plate distortion. An additional point of interest is the fact that the Dodge Pond Test Vehicle used plating with a bend-radius of 12 feet, this stiffening the plating against distortion compared to the radii of 25 feet to 40 feet being considered today.

Distortion of the plating will introduce additional ship's drag, flow noise and mislocation of transducers in the thwartship direction, thus interfering with beamforming. All reasonable attempts at eliminating plate distortion are warranted.

2. The sonar baffle is subjected to severe mechanical loadings. These originate from ship's motion in high seas, the hydrostatic head at the sonar's location and the specification underwater explosion requirement. The hull plating and the point between the hull and sea chest is subjected to high stress levels during these loadings, yet must resist these stresses without failures to insure water tightness of the ship.

3. The large amount of welding on each plate demands a high level of quality control of the welding to insure that minimum number of plates are scrapped and thus to insure a low welding cost. This same high level of quality control insures the adequacy and uniformity of weld strength.

Concern with these problems led TRG to first investigate an automated wire fed arc welding process which would incorporate automatic peening immediately after welding. All attempts at this type of welding yielded weld beads that were too wide for this application. If used, the weld beads for each sea chest would touch the adjacent beads, creating a large continuous welded pattern. Such a weld configuration eliminated too much of the high strength parent material and replaced it with weld bead and weld affected zone. This could not be relied upon to carry the loads.

TRG meanwhile had investigated other forms of welding. Plasma arc gave about the same sized weld zone as MIG and TIG, and ultrasonic welding was not applicable. TRG had built the largest laser welder to date and that device could not supply the energy required for this job.

An investigation into electron beam welding showed that it had the attributes to perform satisfactorily in this project. Among it's advantages are:

1. Excellent penetrating power with the capacity to weld through many times the thicknesses of steel contemplated for this job.

2. A weld zone with a uniform and narrow width. In fact widths equal to 10% of the metal thickness are attainable. This ability of course removes the major source of plate distortion.

3. Completion of a weld in a single pass and automatic control of start and stop of weld in addition to tight control of weld parameters during a weld. This now gave us the ability to control the weld quality.

4. A very small amount of power used to weld. This resulted in a small heat affected zone, again leading to uniform and high weld quality.

5. The ability to weld both similar metals to each other and non-similar metals too. This allowed us a wide range of materials for the hull plating and the sea chest.

6. The possibility of low cost welding, important because of the large number of welds per system.

Among the disadvantages of the process are:

1. Rather limited use to date, but increased usage noticeable for both military and commercial projects. High reliability applications such as fabrication of the Saturn booster and attachment of the XB-70 wings had already been accomplished.

2. The need for a rather high vacuum in which the workpiece must be placed. This results in a higher cost of equipment than conventional welding but simultaneously creates a welding atmosphere that is free of oxygen to a

greater extent and with greater surety than use of an inert gas.

3. The need for a vacuum requires a chamber to contain the vacuum. The workpiece under consideration is too large for all the chambers now in existance.

It was felt at the time that the equipment problem could be solved if the electron beam welding process was satisfactory. Some attempts to get data on the strength of small weldments was then undertaken, with the plan of using the commercial electron beam chambers then available.

Along these lines several courses of action were undertaken. At first, small samples of steel and stainless steel of one half inch thickness were butt-welded together. Then, more elaborate test samples, closer to the expected configuration were fabricated. During this process we learned what was necessary in the way of controlling the fit between the sea chest and the hole in the plate. When we were able to obtain welds that radiographed well, we ran a small test program on the welds produced.

The major problem with the strength of welds in general is low fatigue strength resulting from the cast structure associated with the welding process. Therefore, 18 of 23 test samples were checked for fatigue strength on a rotating beam tester. Three weld test samples were tension tested and a control specimen of each of the base materials, HY-80 and 316 stainless steel was also tested. All tests were performed by a qualified testing laboratory.

The test results were very encouraging. Four of the 23 endurance samples ran in excess of 10,000,000 cycles without failure, with a range of stress of 25,000 psi to 37,500 psi. The weld quality was not too high; evidence of porosity existed whenever a failure occurred. Still the endurance limit for the material was stated by the laboratory as being in excess of 27,000 psi. The tensile test samples revealed a similar picture, with the yield strengths of the welded samples exceeding the yield strength of the stainless steel control sample. The problems of weld quality were felt to be associated with poor set-up and inadequate setting of machine parameters. The set-up problem can be solved by attention, and a weld parameter study can continue this first quantitative effort at investigating electron beam welding. Along these lines, a rather elaborate program ($8000) of welding and testing was proposed to obtain the information on weld quality. It was never approved.

In another direction, an attempt was being made to test transducers and the TRG baffle concept under the shock loading of the UERD test set-up at Norfolk Naval Base. To accomplish this test, 9 sea chests were electron-beam welded to a plate and resonators placed between the sea chests by the stud welding process. Minor failures in the test box and major failures of the transducer resulted from the early tests, but we were able to pass the full series of shots eventually for the electron-beam welding, the resonator attachment and the

transducer. An interesting side line to these results is that the sea chests so stiffened the baffle section that no deformation from the shock resulted. This was not true of even small area panels without sea chests. An indication exists therefore that the large panel (8 feet by 8 feet) with sea chests, has an inherent shock resisting capacity far in excess of the same sized panel without sea chests. This shock hardening feature can be increased by interlocking the sea chests in a hexagonal pattern rather than a square pattern.

For all the above testing, some material choices had to be made. The first cut at material selection resulted in HY-80 for a hull material and 316 stainless steel as a sea chest material. The reasons for HY-80 were high strength, low notch sensitivity, and adequate weldability, while the stainless steel enjoyed high corrosion resistance, high pitting resistance and high ductibility. There is no reason to believe that these materials cannot be welded. Nevertheless it is wrong to base the analysis of the adequacy of electron beam welding for the conformal array program on these materials above. The choice of hull and sea chest materials should be as affected by welding considerations as any other considerations such as strength, ductibility, corrosion resistance, facility of non-destructive testing, cost, etc.

In summary, therefore, the TRG position on welding is as follows:

1. Preliminary investigation has concluded that electron beam welding is the most promising method of assembling the TRG baffle configuration.

2. A series of tests to determine welding parameters and strengths of welds is required to assure the preliminary conclusion.

3. A new investigation into the availability of an electron beam welder with the capability of fabricating an eight foot by eight foot panel should be made because more than one year has gone by since the last such attempt.

4. In the absence of an available electron beam welder, a custom welder to fabricate the baffle should be procured. This machine will be capable of continuing the parametric studies and also be capable of producing test panels, the sea-going panels for the ESS and subsequent models.

Note: TRG has retained Prof. Clyde M. Adams of M.I.T. to give an opinion on the practicability of welding HY-80 steel to 316 stainless steel by the electron beam process. His report will be forwarded as soon as available.

5. TRG Position on the Array-Covering Membrane

A membrane has been proposed to increase the performance of the planar array. The gains from such a construction have been stated as:

1. protection of array structure against the effects of cavitation from high level transducer operation.

2. a reduction in the flow noise sensed by the transducers.

3. a smoothening of the hull resulting in decreased flow resistance and flow noise.

If one assumes that the above statements are accepted as fact there are still disadvantages which must be weighed against the accepted gains to establish whether an increase in cost effectiveness has been accomplished. Some outstanding objections to installing a membrane on a ship are:

1. Increased system initial cost and maintenance cost.

2. A decrease in the effectiveness of the anti-fouling coating. This coating will adhere better to a rigid surface like a metal hull rather than a flexible material such as an elastomeric membrane. If the anti-fouling material can be compounded in the elastomer, and caused to leach out slowly for protection, a limited life for the membrane would still result. Even if this life was as great as some claims, 5 years, the job of removing the membrane and attaching a new one would be costly and time consuming.

3. A susceptibility to increased flow noise due to a section of the membrane being dislodged by bond failure, or relatively mild collision with sea debris.

4. A decreased resistance to the effects of an underwater explosion. At the extremely high pressures in the shock wave of an underwater explosion, the membrane may conceivably fail in the parent material rather than at the bond

5. A decreased margin of safety against sea tightness when the membrane is used as a seal as in the GD/EB MCI approach, because of the unproven nature of the design.

6. A decreased margin of safety against galvanic corrosion when used as an electrical insulator as in the GD/EB MCI approach again due to the unproven nature of this design. In this case a local damage to the membrane which scrapes the aluminum surface treatment would create a catastrophic couple. A small area of the aluminum would become anodic to any unpainted area of the steel hull. This would result in a rapid loss of mass for the aluminum. This condition can rapidly proceed to leaking sea water into a transducer cavity to short out a transducer and then soon thereafter to leaking through the hull with it's attendant dangers.

Only if these above disadvantages still do not outweigh the advantages is it necessary to examine the advantages. As an exercise, assum that is the situation, we must then examine the advantages of the membrane addition.

It has been stated that a membrane would protect the metal hull against damage from transducer cavitation. To allow this statement, there must be evidence that the phenomenon occurs on the sonar being contemplated or in a substantially

similar condition. There is no such evidence. If such evidence existed, one can question whether the high levels of transducer operation which cause cavitation must be tolerated or whether safeguards against that situation are required. Still one step farther, if transducer cavitation must be tolerated, what is the best method of preventing damage to the system of transducers and hull. This is a difficult question to answer. There is evidence that cavitation damage can be reduced by at least three main avenues of design. They are:*

1. Using a metal with a high fatigue endurance limit and a low corrosion rate.

2. Protecting the material with a coating attached by metal spraying or welding overlay. The added material would be identical to the previously mentioned one.

3. Utilizing a bonded elastomeric coating.

Evidently, if cavitation resistance must be added to the sonar structure, there are more possibilities than an elastomeric coating. The method of overlaying or cladding the hull with a cavitation resistant metal layer looks particularly advantageous in this application. A layer of 316 stainless steel or monel is mentioned in the cited article as being effective. In addition one would surely add Hastelloy C. to the list because it is the most corrosion resistant alloy (for sea water) that is known, aside from the fact that it is inherently anti-fouling. When one looks at the advantages

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*How to Protect Materials against Cavitation Damage, Materials in Design Engineering, March 1967.

of such a material in reducing maintenance costs for the hull, it may be indicated even without considerations of cavitation damage resistance.

Returning again to the second stated advantage of an elastomeric membrane, flow noise reduction, we can again question the contention. TRG is working in this area at present and though it feels that a noise reduction is possible, it can not state that a guarantee of noise reduction can be made.

The third contention, that a membrane will smooth a hull, can be easily countered. If the coating is vulcanized to the metal, high temperatures and pressures are used. It is difficult to imagine that deflections in tooling for large pieces and variations in elastomer properties and temperature gradients will leave the elastomer uniform in thickness. Again even if one could apply a uniform thickness of coating, the installation of the array would introduce a dry docking situation where welder's splatter and contact damage are possible. This would in turn demand an unreasonably costly protection or repairs to damage which would compromise the smooth surface. In service, damage from debris, rough seas, fouling, and scraping from anchor chains would soon leave the elastomer surface less smooth than a metal hull can be kept permanently.

To summarize TRG's position with regard to an elastomeric membrane covering the array we state:

1. The use of a membrane to reduce flow noise is a proper subject to investigate.

2. If that investigation indicates that an overall gain in system effectiveness seems possible, then an evaluation of elastomeric materials and other compliant materials such as fiberglass laminate should be undertaken to determine suitability for the application.

3. A large effort in flow noise reduction at this moment is not warranted since it hinders the main line effort of getting a sonar to sea. The program definitely belongs in the product improvement stage of the sonar's life.

4. An effort at investigating the effect on materials near a cavitating transducer should be undertaken to evaluate the severity of the situation. Again this should not hamper the development of the array because many possible solutions exist for the potentially damaging condition.

6. Some Previous Baffle Tests in Water

Some further experimental data which support the position that resonators adequately stiffen a baffle are discussed below. In an early report[1] we show the measurements of the far field pattern of a single transducer element mounted in the end of a drum 6 feet in diameter, operating at 8 kc/s. The radiation pattern comparison is given in Figure III.B-10 one element mounted in the resonator stiffened end and another mounted in the opposite end, without resonators. The pattern on the stiffened side shows an end-fire loss of 6.5 db due to diffraction, while on the side without resonators there is an end-fire loss of 17 db.

In another experimental test[2] we measured the far field patterns of single elements (see Figure III.B-11) and of an array of elements in a resonantly stiffened baffle closely resembling a portion of the proposed sonar keel. Figure III.B-11 shows only the expected 6 db diffraction loss near grazing angles; there are no other losses, which indicates that the baffle has been rigidized.

[1] V. Mangulis, S.Gardner, A.Novick, W.Graham, A Study of Conformal Arrays for Long Range Sonars, Phase Two, TRG-142-TR-2 (1963);(Confidential); Vol. III, Appendix XI; AD 336 258.

[2] J.Lyons, T.DeFilippis, Results of the Dodge Pond Test Program on a 6x12 Element Conformal Array, TRG-142-TN-64-5(1964); (Confidential); AD 377 882.

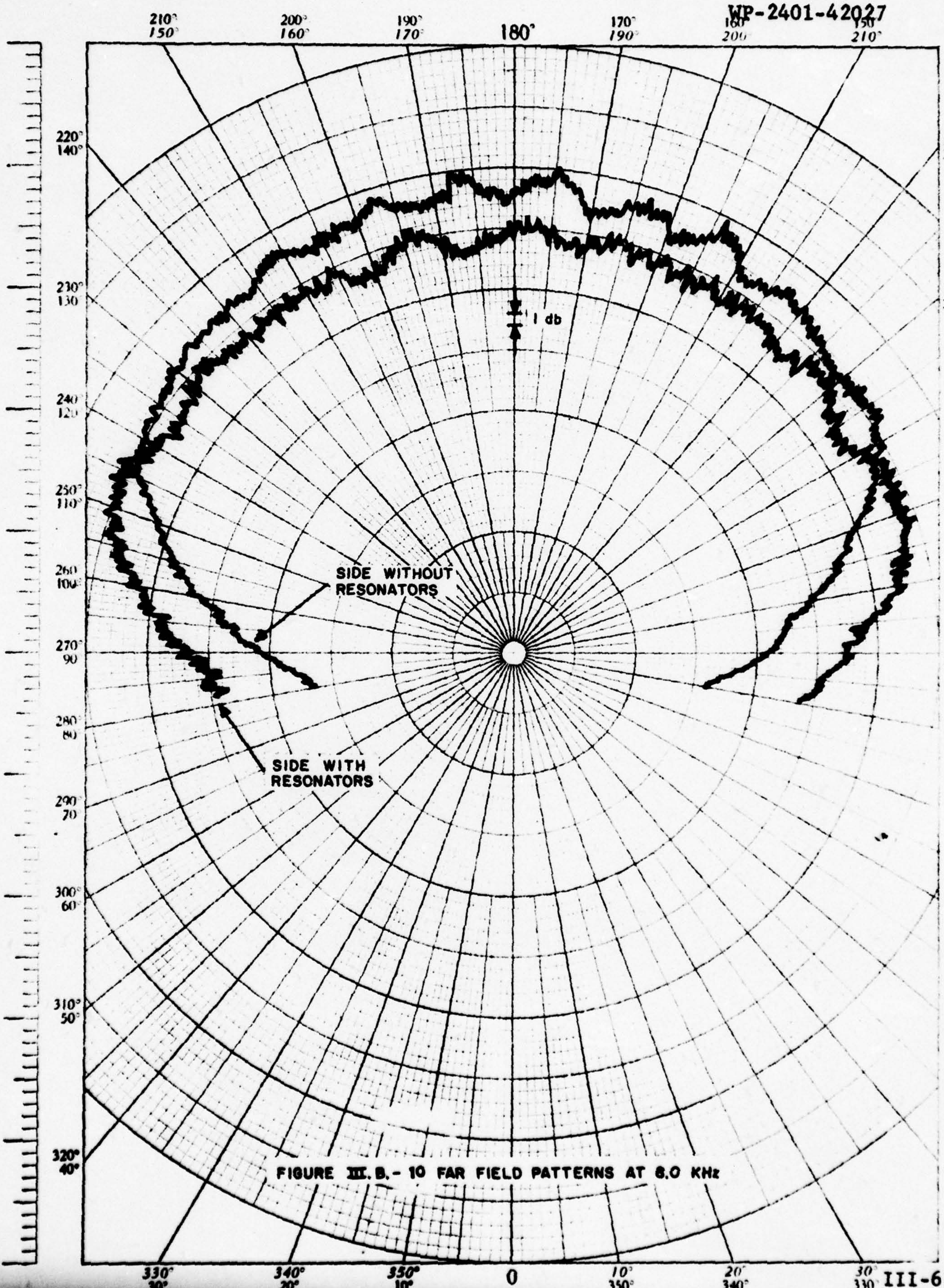

FIGURE III.B.- 10 FAR FIELD PATTERNS AT 8.0 KHz

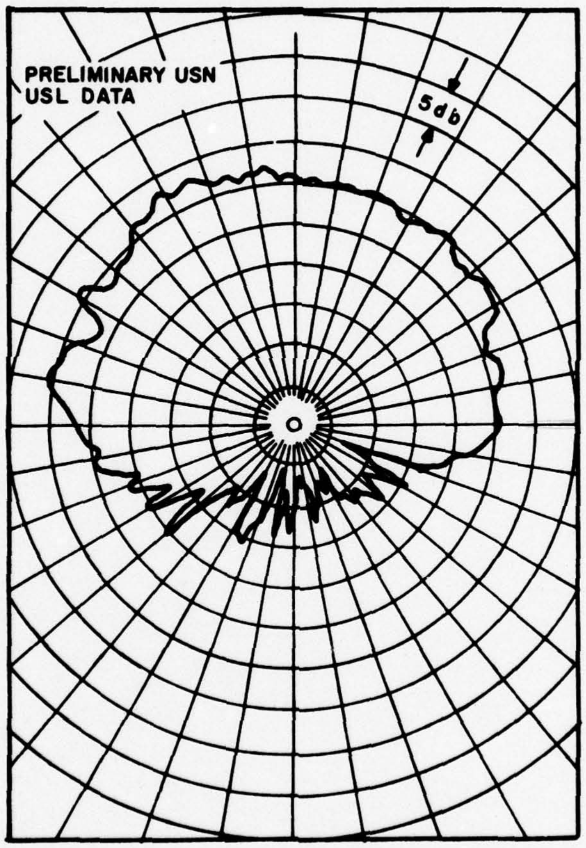

FIGURE III. B.-11 PULSED TRANSMITTING DIRECTIVITY PATTERN AT 4.0 kc FOR A SINGLE ELEMENT MOUNTED IN THE TEST VEHICLE

IV. TRANSDUCER DESIGN

A. Transducer, Load, Radiation Impedance Calculations

For the sparse planar array of circular pistons we have the capability of calculating the radiation load for any piston, beam, and frequency condition in an economic way. We have several computer programs which are now in use at TRG for the computation of radiation impedances of elements of large planar arrays of circular pistons driven with uniform velocity amplitude. The heart of the computation is the same for all of the programs; the difference between programs is in the post-processing of the resulting radiation impedances.

The model used in these calculations assumes that the array and its baffle are contained in an infinite plane and the baffle is considered infinitely rigid. These assumptions are reasonable since the diffraction effects due to baffle finiteness and array curvature upon radiation impedance are negligible. These diffraction effects are discussed in Section II.A and are shown to be tolerable if neglected. As for baffle rigidity, we have demonstrated experimentally that we have achieved adequate baffle rigidity in our Dodge Pond experiments and recently in scale model measurements. The mechanism by which rigidity is obtained is theoretically explained in Section III. Furthermore, transient effects due to initial propagation delay cause radiation impedance changes of approximately 50% and since the elements of the array are velocity controlled this amount of variation is not a problem; transient effects are discussed in Section II.B.

The heart of the radiation impedance program is the calculation of radiation impedance of each piston in the array. If the velocity distribution is given, then the radiation impedance is defined.

In our calculation, we consider all individual element velocities to be of equal magnitude, and to have phases which make all pressures add constructively in the steering direction. The magnitudes of the velocities are determined from consideration of cavitation after all impedances are computed. For the computation, unit magnitude is used.

Since we know the velocity, we can use the mutual impedance for circular pistons on an infinite, rigid, plane baffle; consequently, it seems like a straight forward procedure to compute the radiation impedance for all pistons for each beam, and computationally it is true. However, this method requires quite a bit of computer time. To compute all the impedances by brute force at one steering angle requires approximately 0.12 hours of computation for an array of about 2500 pistons.

We can, however, take advantage of the fact that the velocity magnitudes are all equal, and that the columns of the array are uniformly spaced. By taking advantage of this symmetry we obtain a large saving in computation time. The time required by this method was found to be .01 hours per beam for 2500 pistons, a better than 10 to 1 savings in computational cost.

In addition we single out the locations and impedances of the following pistons: those with maximum resistance, reactance magnitude, and phase; minimum resistance, and reactance; average resistance and average reactance. We also compute the velocity which causes cavitation, the acoustic power output, the directivity index, source level, self impedance, and characteristic impedance.

Numerous calculations of radiation impedances for different array configurations have been performed over the years.[1,2] The ranges of radiation impedance values encountered indicate that it is possible to design a sparese array with circular piston elements so as to have velocity control.

B. Determination of an Acceptable Rocking Impedance Level

The acceptable rocking impedance level is much less of a problem with 5" diameter sparse circular elements than with 7" square dense elements. In an analysis to arrive at this acceptable level we must consider the effects on the far-field radiation of the array due to this rocking motion of the transducers. To this end a quantity known as Wobble Impedance is defined as the ratio of moment on the piston

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[1] V. Mangulis, A. Kane, Some Acoustic Calculations for a Large Array, TRG-011-Memo-64-2 (1964); Confidential.

[2] A. Kane, Computation and Collation of Radiation Impedances, TRG-023-TM-66-33 (1966).

face to the angular velocity of the face. Obviously, there is some acoustic radiation due to the rocking.[3] We estimate these effects for an infinite planar array of circular pistons, and while we admit that the numbers obtained are for an infinite array only and are estimates, they may be useful in the design of sonar transducers to be used in a finite array.

To find the moment on the piston, we consider the near field pressure in an infinite array of circular pistons.[4] The moment is found by integrating the pressure over the face of the piston. The ratio of this moment and the angular velocity is the rocking impedance.

It should be noted that the radiation impedances for the infinite array are in very close agreement with those for a large finite array for angles not too near end fire.[4] Therefore we can assume that for the finite array we can use the rocking evaluated at some angle near end fire and get reasonable estimates. The results of these calculations for a 1 db degradation at 15° from end fire are: 995 in.-lb.-sec./rad. for 3 kc/sec. and 313 in.-lb.-sec./rad. for 2 kc/sec.

If we have a transducer with a wobble impedance higher than these values over the 2-3 kc band, we assume that the ratio of wobble velocity to longitudinal velocity is small enough so that the far-field pattern will not be degraded by more than one decibel at end fire. Note that

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[3] V. Mangulis, Acoustic Radiation from a Wobbling Piston, J. Acoust. Soc. Am. 40, 349 (1966).

[4] V. Mangulis, Near Field Pressure for an Infinite Phased Array of Circular Pistons, TRG-023-TN-66-18 (1966); to appear in J. Acoust. Soc. Am., Feb. 1967 issue.

this degradation includes the directivity of the piston in the longitudinal mode. If this directivity is not included, the acceptable wobble impedance levels are lower than those listed above. For the purposes of this study, we use the more stringent constraints.

C. Wobble Impedance of the Transducer Element

The wobble impedance of a transducer element is defined as the complex ratio of torque at the face of the element to angular velocity of the face.[3] The determination of this quantity is carried out using a two degree of freedom model of the transducer - "Timoshenko Beam Theory." This model is used because it includes effects of pure bending, shear deformation and rotary inertia of the several sections of a transducer, and it couples these motions together.

Since the transducer element does not have constant cross-section geometry along its entire length, and since the solution to the "Timoshenko Beam" equation is not conveniently expressible in a suitable closed form, a finite-difference equation technique is used to obtain a solution.

The "beam" is divided into segments (each being small compared to the flexural and shear wavelengths at the frequencies of interest), and in each segment the differential equations are represented by finite difference equations. Obviously, as the number of divisions increases, the approximation gets better, and in the limit, as the length of each segment approaches zero, the difference and differential equations are the same. A detailed description of the method of setting up the

difference equations is given in Ref. 5.

The difference equations are represented by an electrical circuit analog. Each section of the beam is analogous to a four-port electrical network-two input and two output ports. One port at the input side represents bending, and the other port represents shear deformation. In the "Timoshenko model," these deformations are coupled within the section. The four-port representation permits matching at the boundaries of the following quantities:

a) y-deflection (time derivative),

b) slope (time derivative),

c) moment,

and d) shear.

Note that since sinusoidal time variation is assumed, time differentiation is equivalent to multiplication by ix(radian frequency). These four boundary conditions characterize the two degree of freedom system.

The circuit for each section is represented by a 4 x 4 transfer matrix, and the transfer matrices are cascaded to form the overall transducer transfer matrix. The wobble impedance is obtained from this matrix by applying the following boundary conditions:

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[5] Barnoski, R.L., The Use of Passive Element Electrical Analogs in Numerically Calculating the Response Characteristics of Beams, IEEE Transactions on Aerospace - Support Conference Proceedings, AS-1, pp. 474-492 (1963).

a) moment at rear end is zero,

b) shear at rear end is zero,

and c) ratio of displacement-to-shear at front end is input (usually a very large number, so that the shear is effectively zero at the front end.)

These boundary conditions are sufficient to determine the ratio of moment to angular velocity (time derivative of slope) at the front face of the transducer.

D. Initial Transducer Design (Considering Rocking and Velocity Control)

Since the rocking impedance is the characteristic of present concern, the initial design utilized a ceramic stack which performed well in the longitudinal mode. This stack was the one used in the C.P. 1.1 element.

A 5.85 kg rear mass was used with the above stack and the front mass was chasen such that the voltage across the stack was minimized at 2.5 kc/s for a given acoustic drive. This resulted in a 3.4 kilogram front mass. The element configuration, as seen in Figure IV.D.-1 does not include a tension bolt and it utilizes a cylindrical section instead of a conical section for the front mass. This modification does not materially alter the longitudinal or rocking behavior of the element.

A longitudinal evaluation of this transducer element was made by subjecting the design element computationally to the extreme acoustic loads encounted in 21 sample beams for an array of 12 x 229 five inch elements. These extreme loads include those which possess maximum

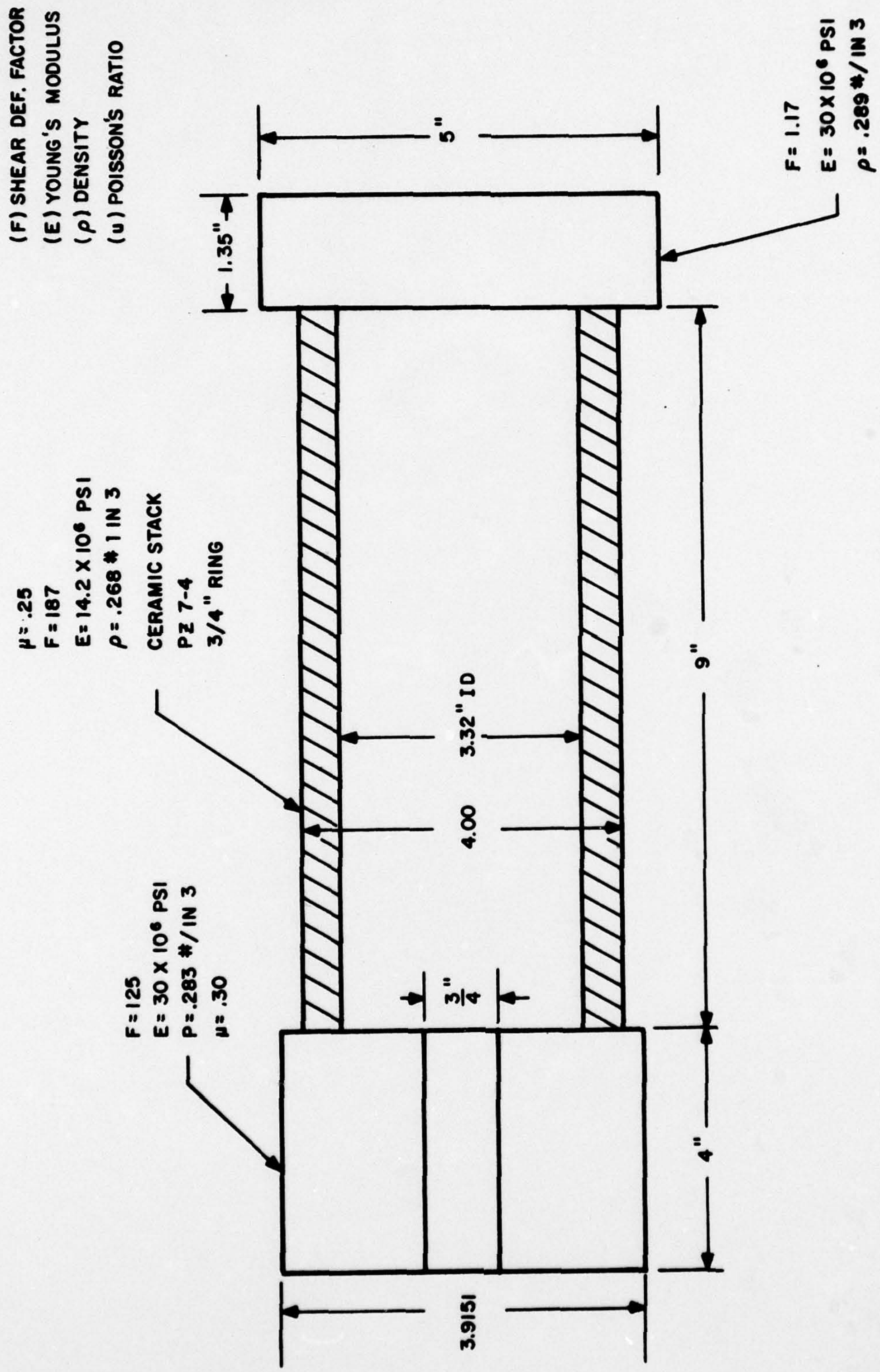

FIGURE IV.D.-1 5" ROUND TRANSDUCER ELEMENT

resistance, inductance, impedance, and phase angle; minimum resistance and inductance.

The results of this computation is the VA, input power, input phase angle, efficiency, and stack voltage for each extreme load. However since it is generally agreed that the VA required to drive the transducer to cavitation is a primary problem area, the maximum VA required is plotted in Figure IV.D.-2 versus frequency. Also plotted is another variable of interest, stack voltage. This evaluation was run for three 300 c/s bands low, mid, and high, centered at 215, 2.5 and 2.85 kc/s respectively. In each band the input tuning was held constant. Note from this plot that in this element the 600 VA maximum was not exceeded and the electric stress of 5 volt/mil was not exceeded. The switching of the input tuning network is only necessary because of the requirement of limiting the maximum VA drive required of the driver amplifier to be less than 600 VA when transmitting. However during reception because of the large Z_{oc} (mechanical output impedance) bandwidth of this element, the total sonar band can be received by use of only the mid band input tuning network.

After examining the results of the VA plot it was included that a minimization of the stack voltage at 2.4 instead 2.5 kc/s would have reduced the VA drive required toward the low band.

During the rocking impedance analysis of the element several variations of this element which have the identical longitudinal behavior were examined in regard to their rocking resonance behavior. This involved

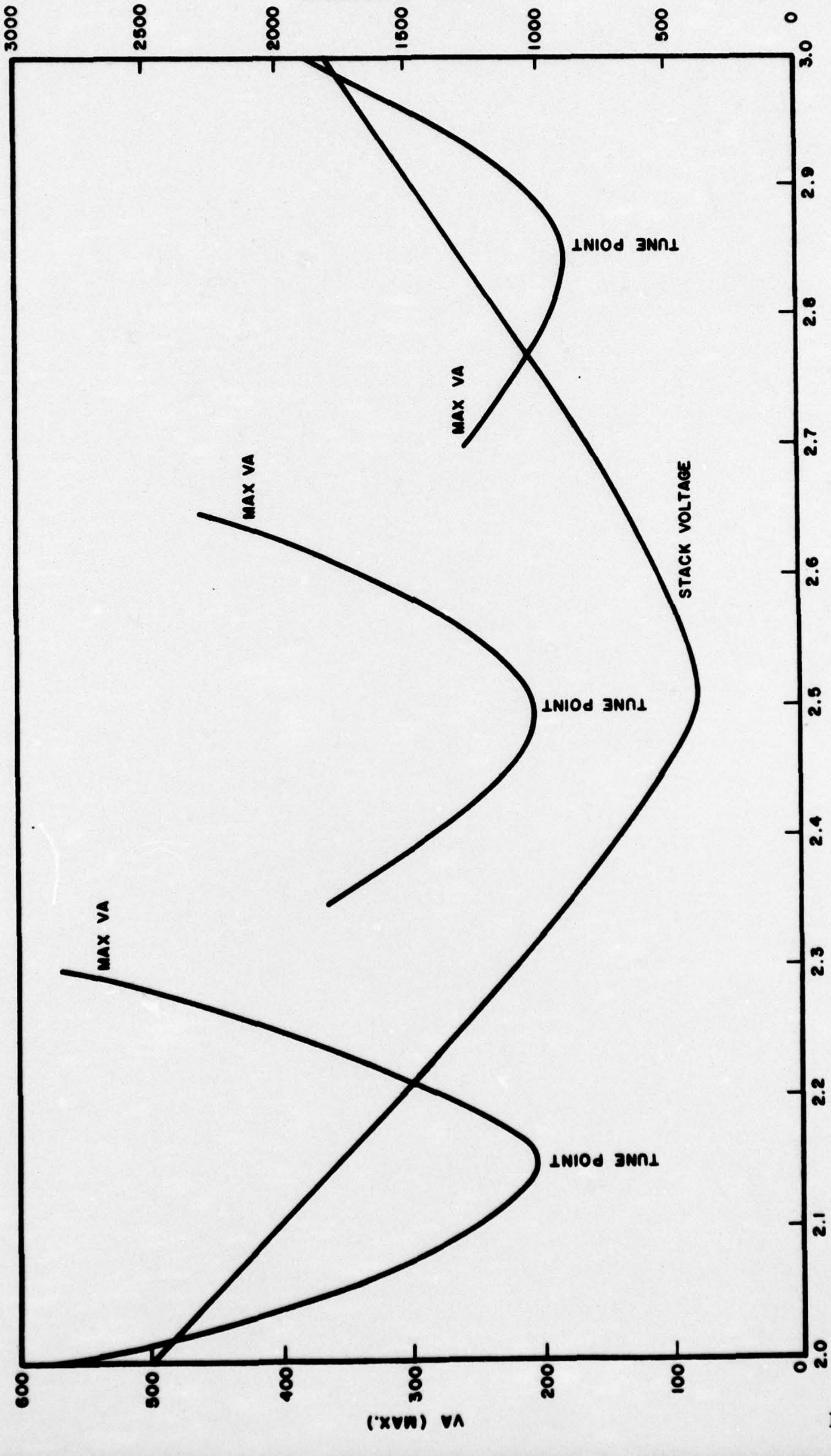

FIGURE IV.D-2 VA AND STACK VOLTAGE VS. FREQ. FOR 5" ROUND ELEMENT

a study of the rocking impedance variations with changes in O.D. of the ceramic stack for constant cross section area of the stack and changes in rotary inertia of the back mass for a constant mass.

Figure IV.D.-1 shows the element configuration which had the largest rocking impedance bandwidth as is indicated in Figure IV.D-3. This bandwidth is a function of the acceptable wobble impedance level discussed in Section IV-B.

It should be noted that in these rocking calculations we used the same shear factors that G.D. use in the rocking analysis of the C.P. 1.1 element so that the results may be compared. The shear factor for a solid cylinder is 10/9 and not 1.17 as G.D. used.

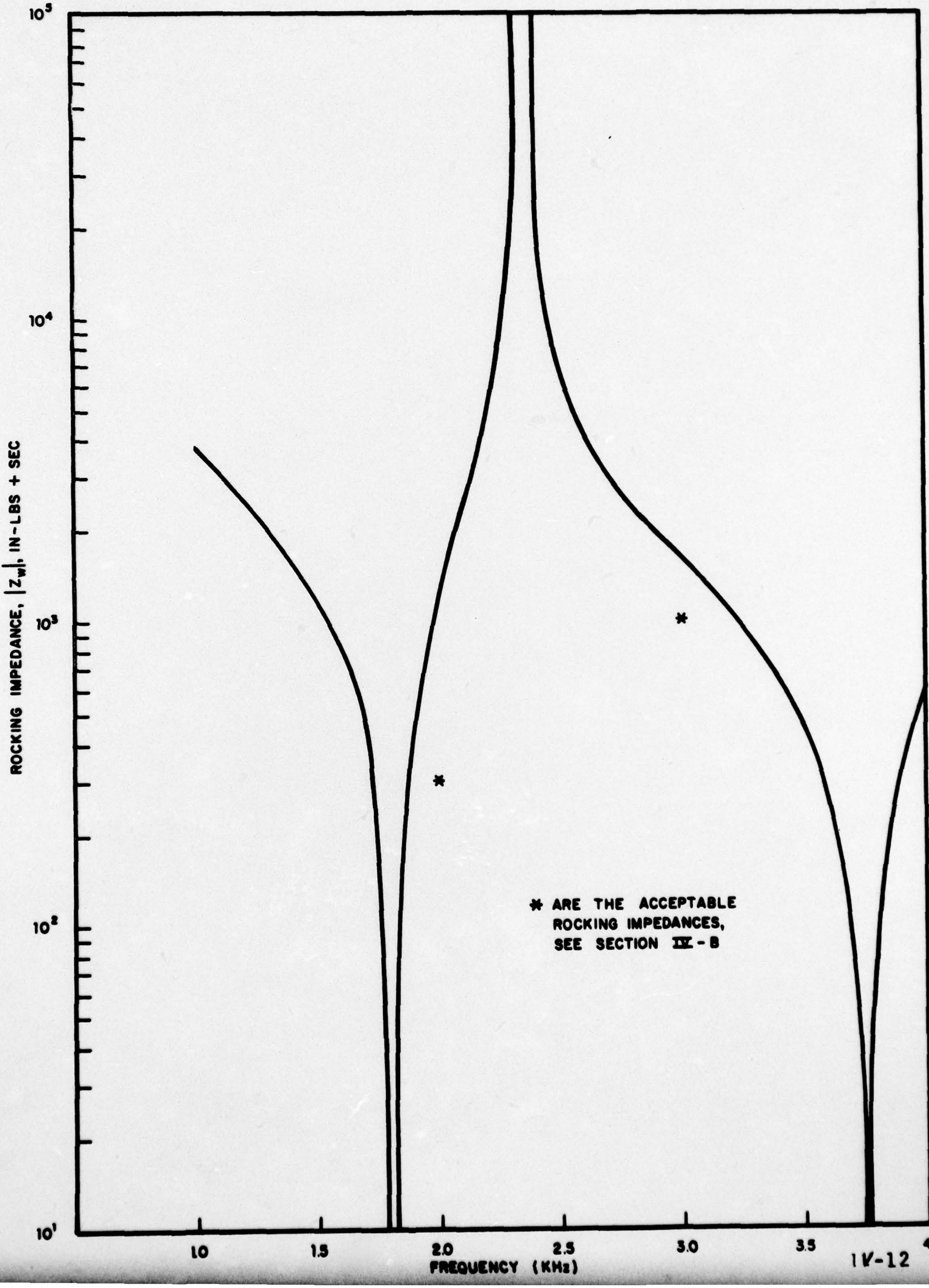

FIGURE IV D-3. ROCKING IMPEDANCE OF THE 5" ROUND ELEMENT

V. NAVAL ARCHITECTURE

The manner in which the conformal/planar sonar array baffle is stiffened for use with a sparse transducer array, or the choice of sparse or dense transducer packing (array factor), affects the naval architectural characteristics of the vessel upon which the array may be installed in several ways. The basic effects have to do with the weight, volume and power requirements of the particular sonar array design under consideration. These factors change the total vessel's weight (and buoyancy for equilibrium) both in magnitude and distribution, and therefore its resistance, stability, longitudinal bending strength, seaworthiness and possibly its maneuverability. The possibility exists, but is considered not likely, that the type of baffle, or choice of array factor, may cause a change in the hull form configuration, or shape of the hull including the sonar array. Such a change would also affect the vessel's weight and buoyancy relationship as well as the resistance, stability, etc. It would also change the underwater flow conditions which determine the bubble sweepdown conditions.

With regard to the proposed ESS installation aboard the USS SPOKANE, all of the comments given in Section 6 of TRG's report of March 1966[1] still apply. In addition to this

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[1] Recommendations for Installation of ESS on the USS SPOKANE, TRG Document No. B033-42001, March 17, 1966.

material a preliminary design and cost estimate of the keelster has been prepared by M. Rosenblatt & Son, Inc.[2]. This report treated in detail the design for the general approach of a sparse array with resonant baffle stiffening within the keelster type hull configuration. The sonar keel approach, also presented in Reference [1], has been model tested at DTMB for resistance and flow characteristics [3]. These tests showed the relatively small increase in effective horsepower due to the addition of the sonar keel appendage to the SPOKANE. The proper interpretation of the circulating flow channel data is still under consideration, but it does appear that the flow conditions (with respect to bubble sweep-down) are as good or possibly improved over that of other hull configurations (on the SPOKANE) except for those which incorporate a slot, or open space between the hull and array.

The buoyance increase due to the sonar keel on the SPOKANE is 320 tons S.W. The added weights due to the ESS installation are estimated in Table V-1 on the next page. Also shown on the same basis for comparison is the weight estimate detailed by GD/EB (4/1/67), for the keelster hull configuration and with mass-controlled-interstice (MCI) baffle stiffening. The difference in transducer, baffle and structural weight groups are seen. The sub-total weight of items directly associated with the sonar keel

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[2] Preliminary Design and Cost Estimate for Installing ESS/CP Sonar Semi-Keel on CLAA-120, USS SPOKANE, M. Rosenblatt & Son, Inc. Report No. P&CD-1712-2, June 1966.

[3] Lin, Alan C.M. and Grant, Jerald W., Flow Characteristics and Prediction of Effective Horsepower for the Conformal/Planar Array Sonar Program - USS SPOKANE with TRG Designed Sonar Keel - Represented by TMB Model 3542-3, David Taylor Model Basin Hydromechanics Laboratory Test Report C-108-H-14, January 1967, CONFIDENTIAL

TABLE V-1
SONAR SYSTEM WEIGHT SUMMARY
ESS INSTALLATION ON USS SPOKANE

Item	Weight (Long Tons) TRG Sonar Keel Resonant Baffle Stiffening	Weight (Long Tons) GD/EB Keelster MCI (alum. block) (4/1/67)	
Transducers	60.8*	43.0	
Transducers Support Structure (Baffle)	105.3*	86.8	= 101.5
Array Support Framing		14.7	
Plating below array	111.0**	30.7	= 197.8
Longitudinal bulkheads		26.4	
Bulkheads		15.0	
Bottom Fairing		20.2	
Aft fairing		21.9	
Structure above array		37.2	
Forward fairing		19.1	
Stbd. Shell in way of ballast tank		27.3	
Machinery***	5.0**	0.0	
Electrical***	2.2**	0.0	
Ventilation***	6.0**	0.0	
Foam Fillers	2.0	4.0	
Sub-Total	292.3	346.3	
Cables, transducers to sonar equipt. room	15.0	15.0	
Sonar equipment	63.0	63.0	
Cables to power generation room	6.0	6.0	
Power generation equipt.	76.0	76.0	
Total	452.3	506.3	

*Based on weight breakdown shown on Table V-2 on next page.

**From "Preliminary Cost Estimate, C/P Array CLAA120-USS SPOKANE", M. Roseblatt & Son, Inc., March 14, 1966, included in Section 6 of Reference [1]. The validity of these estimates is borne out by the detailed estimate for the keelster in Ref. [2].

***Includes only ship service functions such as bilge systems, lighting, etc.

TABLE V-2

WEIGHT BREAKDOWN

TRANSDUCER AND RESONANT STIFFENED BAFFLE FOR ESS INSTALLATION

(Weights are given in pounds per transducer unit)

(Note: number of transducer units on one side is 12 rows x 225 columns = 2700 units.)

TRANSDUCER

Front mass	13.2
Rear mass	17.6
Ceramic stack	10.0
Shell (enclosure)	9.0
Elastomeric face	0.6
	50.4 lbs.

$$\left(\times \frac{2700\text{ units}}{2240\text{ lbs/ton}} = 60.8\text{ tons}\right)$$

BAFFLE

Plating (3/4" thick)	8.1
Sea chest	61.0
Resonators	11.2
Rear cover	4.5
Rear cover hardware	1.0
Shock retainer	1.5
	87.3 lbs.

$$\left(\times \frac{2700}{2240} = 105.3\text{ tons}\right)$$

itself is 292.3 tons and indicates that the keel appendage is close to neutrally buoyant. The additional weights of cables, sonar equipment and power generation equipment is taken to be the same as estimated by GD/EB since they should not change between the two arrangements (keel vs. keelster, resonant vs. MCI stiffening). Though a current detailed estimate, both for the items shown on Table V-1 as well as for other ship modification items, has not been prepared, it does appear that the effects of weight and buoyancy, for a sonar keel approach with sparse array and resonant stiffened baffle, will not create seriously adverse problems for ESS installation on the SPOKANE. This is based on the above rough estimate and on TRG's detailed contract design and specifications for construction and installation of a sonar keel on the DD709[4]. Further calculations should be based on detailed information regarding the final status of armament and ammunitation weights, fuel and crew requirements and other specific ESS information. Consideration should then be given to the basic stability and strength questions.

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[4] Specifications for the Installation of CONTACT Sonar System, Modifications to the USS HUGH PURVIS (DD709) (and attached drawings and calculations), M. Rosenblatt & Son, Inc. No. 27576, May 1965, CONFIDENTIAL

VI. CONCLUSIONS

The array configuration we propose uses an air backed baffle around the transducer elements, which is part of the supporting structure for the elements. The air backing of the baffle is essential to the use of the principle of resonant stiffening of the baffle. The stiffened baffle makes possible excellent control of beam patterns (the only vibrating surfaces are the elements), and tractability in computation of radiation impedances. The resonant stiffening also suppresses structure borne vibrations. The sonar keel structure provides a water tight housing in which cables, switches, junction boxes and other components of the sonar can be located. The housing displaces, approximately, the weight of the sonar. It provides access to all the transducer elements which can be replaced with the ship in the water. The geometry is compatible with the use of a dome should improvements in dome design make the use of a dome desirable, and justify the added cost. A considerable sacrifice in ease of maintenance would result, however. The geometry is also compatible with use of a boot adhered to the housing which might provide noise reduction without so great a penalty in maintainability, and at a lesser cost.

The array factor in the proposed sparse array configuration is about 31%. If we compare the cavitation limited power outputs of a sparse array (31% array factor) and a relatively dense array (65% array factor), we find that the power output is larger for the dense

array by less than 0.5 db near end fire, and by about 2 db near broadside.[1] In other words, near end fire, where the array performance is critical, a sparse array will perform just as well as the more expensive and heavier dense array. It seems indisputable that the rocking effects are less severe for the small elements in the sparse array than for the large elements in a dense array.

Our baffle design, resonantly stiffened, has been tested experimentally in water several times. The proposed sonar keel is load bearing and shock-resistant. The resonant stiffening also suppresses structure borne and flow excited vibrations of the baffle.

We believe that the experimental tests and theoretical analyses which we have performed are sufficient to prove that the sparse array in a baffle with resonant stiffening offers the best weapon system performance for a fixed amount of money. We do not think that any new major tests or calculations are necessary, and we believe that a prototype C/P array could be built immediately.

Some questions have been raised about the applicability of measurements in air to results in water. As explained in Section I, we are convinced that the measurements are indeed applicable. Nevertheless, to resolve any remaining doubts, one might repeat the set of measurements in water. However, the particular measurements in air are not really essential to our argument that we have sufficient proof that the resonantly stiffened baffle with a sparse array yields a superior weapon system performance.

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[1] Recommendations for Installation of ESS on the USS SPOKANE, TRG report (March 1966).